INVESTIGATION OF SECONDARY LOOP SUPERMARKET REFRIGERATION SYSTEMS

CONSULTANT REPORT

Prepared For:
California Energy Commission
Public Interest Energy Research Program

Prepared By:
Southern California Edison
Foster-Miller, Inc.

March 2004
500-04-013

Prepared By:

Southern California Edison, RTTC
P.O. Box 6400
Rancho Cucamonga, CA 91729

Ramin T. Faramarzi, P.E.
Principal Investigator

Foster-Miller, Inc.
350 Second Avenue
Waltham, MA 02451

David H. Walker, P.E.
Principal Investigator

Contract No. 500-98-039

Prepared For:

California Energy Commission

Chris Scruton,
Project Manager

Nancy Jenkins,
Program Area Lead
Buildings End-Use Energy Efficiency

Terry Surles,
PIER Program Manager

Robert L. Therkelsen,
Executive Director

pier
PUBLIC INTEREST ENERGY RESEARCH
"Research Powers the Future"

Acknowledgements

The authors would like to thank Chris Scruton and Nancy Jenkins of the California Energy Commission for their help and guidance throughout this project. We would also like to recognize the outstanding work and cooperation that we received from personnel at several organizations:

Safeway, Inc.	Rob Uhl
Hill PHOENIX, Inc	Henry Pellerin Scott Martin Georgi Kazachki David Hinde Yakov Arshansky Ken Mongrain
Southern California Edison's RTTC	Bruce Coburn Rafik Sarhadian

Authors:

Ramin T. Faramarzi, P.E. - Southern California Edison, RTTC
David H. Walker, P.E. - Foster-Miller, Inc.

Preface

The Public Interest Energy Research (PIER) Program supports public interest energy research and development that will help improve the quality of life in California by bringing environmentally safe, affordable, and reliable energy services and products to the marketplace.

The PIER Program, managed by the California Energy Commission (Commission), annually awards up to $62 million to conduct the most promising public interest energy research by partnering with Research, Development, and Demonstration (RD&D) organizations, including individuals, businesses, utilities, and public or private research institutions.

PIER funding efforts are focused on the following six RD&D program areas:

- Buildings End-Use Energy Efficiency
- Industrial/Agricultural/Water End-Use Energy Efficiency
- Renewable Energy
- Environmentally-Preferred Advanced Generation
- Energy-Related Environmental Research
- Strategic Energy Research.

What follows is the final report for the Investigation of Secondary Loop Supermarket Refrigeration Systems, Contract No. 500-98-039, conducted by Southern California Edison RTTC. The report is entitled Investigation of Secondary Loop Supermarket Refrigeration Systems. This project contributes to the Buildings End-Use Energy Efficiency program.

For more information on the PIER Program, please visit the Commission's Web site at: http://www.energy.ca.gov/research/index.html or contact the Commission's Publications Unit at 916-654-5200.

Table of Contents

List of Tables

List of Figures

Executive Summary

Background

The largest single use of energy in a supermarket is refrigeration, accounting for half of total energy use, on the order of 2-3 million kWh annually for a 35,000 ft^2 store. Refrigeration is typically provided by direct expansion air-refrigerant coils that are located in the display cases and walk-in coolers. The compressors are located in a machine room in a remote part of the store, such as in the back room area or on the roof. As a result of using this layout, the amount of refrigerant needed to charge a supermarket refrigeration system is very large. A typical store will require 3,000- 5,000 lb of refrigerant. The large amount of piping and fittings used in supermarket refrigeration leads to significant refrigerant leakage. On average, a supermarket refrigeration system can be expected to lose 30 – 50% of the total charge annually.

With increased concern about the impact of refrigerant leakage on global warming, new supermarket system types requiring significantly less refrigerant charge are now being considered. Of these advanced systems, the secondary loop refrigeration system employs central mechanical systems and can reduce the total refrigerant charge to approximately 300 – 500 lb. The anticipated refrigerant leak rate for the secondary loop refrigeration system is much less than that of the direct-expansion systems, because all refrigerant is contained within the chiller system. Refrigeration is provided to display cases by a secondary fluid pumped between the cases and the chiller system. Secondary loop systems, however, have tended to use more energy than multiplex systems.

The overall goal of this project was to determine if an advanced, high-efficiency secondary loop refrigeration system could have a lower operating cost than a state-of-the-art multiplex refrigeration system, considering both energy and refrigerant use. This would make the environmentally benign secondary loop system economically attractive.

Project Approach

Southern California Edison and its subcontractor, Foster-Miller, Inc., obtained support from Safeway, Inc. to provide a supermarket where an advanced secondary loop refrigeration system could be installed and tested. After design and analysis work was performed, a specification for the advanced secondary loop system was prepared and several refrigeration manufacturers were asked to propose and bid on its construction. Hill PHOENIX, Inc. of Conyers, GA was selected as the system manufacturer and installing contractor based on their response to this solicitation.

The advanced secondary loop refrigeration system was installed and instrumented for evaluation of its performance. At that same time, a second supermarket operated by Safeway was identified that employed a state-of-the-art multiplex refrigeration system. The multiplex system was also instrumented for performance measurement and both sites were monitored for approximately 9 months.

Project Objectives and Results

The specific technical objectives of the project were to build and test a secondary loop refrigeration system, which:

- *Consumes approximately 14% less electricity than a state-of-the-art multiplex refrigeration system (baseline system) installed in a comparable store.*

The modeling results for the secondary loop refrigeration system designed for this project predicted annual energy savings of 145,894 kWh, or 14.9%, when compared with a state-of-the-art multiplex refrigeration system with air-cooled condensing. The multiplex system with an air cooled condenser is the most common configuration found in supermarkets.

In the field test, the secondary loop system was compared to a multiplex system with evaporative condensing, which is more efficient than air cooled condensing. This was appropriate because the secondary loop system also used evaporative condensing. Modeling results predicted annual savings of 6,130 kWh, or just under 1% for this comparison, but the actual savings achieved by the secondary loop refrigeration system were 37,266 kWh/yr, or 4.9%. This suggests the benefit of the secondary loop system as tested over an air cooled multiplex system would have been greater than 14.9%.

The savings achieved by the secondary loop refrigeration system can be attributed to energy-saving features incorporated in its design. The annual savings achieved by each of these features compared to a more conventional secondary loop refrigeration system were:

- Multiple, parallel brine pumps – Estimated annual savings versus single large pumps are 99,718 kWh.
- Subcooling from warm brine defrost – Estimated annual savings versus no subcooling are 49,570 kWh.

Additional savings were also achieved by the use of a minimum difference between the display case discharge air and refrigeration saturated suction temperatures, the use of a low-viscosity secondary fluid, and evaporative condensing as noted previously.

- *Has a refrigerant charge that is ten times less (less than 500 lbs.) than the baseline system.*

The advanced secondary loop refrigeration system as tested had an initial refrigerant charge of 1,400 lb., which is considerably larger than the stated goal. The reason for the larger refrigerant charge is that the secondary loop system also had heat reclaim capabilities for both hot water and space heating. The added charge was needed for piping and heat exchangers associated with heat reclaim. Heat reclaim is of great value to the operation of the supermarket since it is capable of displacing all energy use associated with hot water heating and space heating for the store (Estimated to be approximately 7% of the total store energy use). Without the heat reclaim equipment the refrigerant charge would have been approximately 500 lbs. Further design enhancements such as including the heat recovery equipment within the factory-made chiller equipment room could result in a reduced overall refrigerant charge.

- *Loses annually no more than 15% of its refrigerant charge due to leakage.*

Service records allowed refrigerant leak rates for the 2 stores to be calculated, using the methods provided by the USEPA. These calculations showed that the refrigerant leak rate for the secondary loop store was on the order of 14.8%/yr, or 206 lb/yr. The multiplex system was estimated to have a leak rate of only 12.4%/yr, but because of its much larger charge this amounted to 370 lb/yr. The refrigerant data for the multiplex store were very limited stemming from a change in maintenance contractors. It is likely that a full year of data would show a larger loss of refrigerant for the multiplex refrigeration system. Over time, and as equipment is serviced, the more scattered refrigeration apparatus of the multiplex system will probably begin to leak more, which would increase the relative benefit of the more centralized secondary loop system.

The ability of the 2 refrigeration systems to maintain product storage temperature was also assessed. The comparison for single-deck meat cases showed that the multiplex and secondary loop systems maintained the product at acceptable temperature levels. The case associated with the multiplex system had a lower and more uniform product temperature than was seen for the case operating in the secondary loop system. The multiplex display case had to operate a much lower rack SST (2.3°F for multiplex vs. 14.1°F for secondary loop) in order to achieve this condition. For the multi-deck produce cases, the multiplex and secondary loop systems operated at similar rack SST values. The resulting average product temperature was approximately the same for both systems, but the product temperature of the multi-deck case in the secondary loop system was more uniform in value.

Conclusion

Based on this research, we can conclude that secondary loop refrigeration systems are a viable option for supermarket refrigeration because:

- With efficient design the energy consumption of a secondary loop system can be less than a multiplex system. The project system consumed 4.9% less electricity than the baseline multiplex system.
- Refrigerant use is reduced in proportion to the quantities of refrigerant, piping and fittings employed. Refrigerant leaks are difficult to find and repair, and after years of repairs and modifications the multiplex system will probably leak more than the compact and centralized secondary loop system.
- The somewhat higher first cost for the secondary loop system will probably be mitigated over time by lower operating and maintenance costs.

Benefits to California

This project contributed to the PIER program objective of reducing the environmental costs of California's electrical system, by developing an alternative refrigeration system which uses significantly less refrigerant than conventional systems. It also contributed to the PIER program objective of improving energy value of California's electricity by lowering electrical consumption of supermarket secondary loop refrigeration systems.

Abstract

Present multiplex supermarket refrigeration systems consume approximately 1 to 1.5 million kWh/yr. and may lose annually as much as half of the 3,000 to 5,000 lb system refrigerant charge. The secondary loop refrigeration system employs fluid loops and a central chiller to provide refrigeration to the supermarket display cases requiring just 300 to 500 lb of refrigerant for operation. An advanced secondary loop system was investigated, taking advantage of energy-saving features to reduce energy consumption. Modeling of the advanced system indicated energy savings of 14.9 and 0.3% versus multiplex systems with air-cooled and evaporative condensing, respectively. Results of a field test comparison of the secondary loop system, a multiplex refrigeration system with evaporative condensing developed energy savings of 4.9% for the secondary loop refrigeration system. Savings produced by the secondary loop refrigeration can be attributed to: minimum temperature differences between display case discharge air and saturated suction temperatures; the use of multiple parallel brine pumps; low-viscosity secondary fluid; and subcooling produced by the warm fluid defrost system. An evaluation of product temperature for similar display cases at each site was conducted. The secondary loop refrigeration system maintained the same or lower product temperatures than the product temperatures achieved by the multiplex refrigeration system.

1.0 Introduction

1.1 Present Supermarket Refrigeration Systems

One of the largest uses of energy in supermarkets is for refrigeration. Most of the product sold is perishable and must be kept refrigerated during storage and display. Typical energy consumption for supermarket refrigeration is on the order of half of the store's total (Total store energy use is on the order of 2-3 million kWh annually for a 35,000 ft^2 store). Compressors and condensers account for 30-35%. The remainder is consumed by the display and storage cooler fans, display case lighting, and for anti-sweat heaters used to prevent condensate from forming on doors and outside surfaces of display cases.

Figure 1 shows the layout of a typical US supermarket refrigeration system. Direct expansion air/refrigerant coils, located in the display cases and walk in coolers, provide refrigeration. The compressors are located in a machine room in a remote part of the store, either in the back room area or on the roof. The system condensers are located either in the machine room, or more likely, on the roof above the machine room. Piping is provided for refrigerant supply and return between the machine room and the refrigerated fixtures.

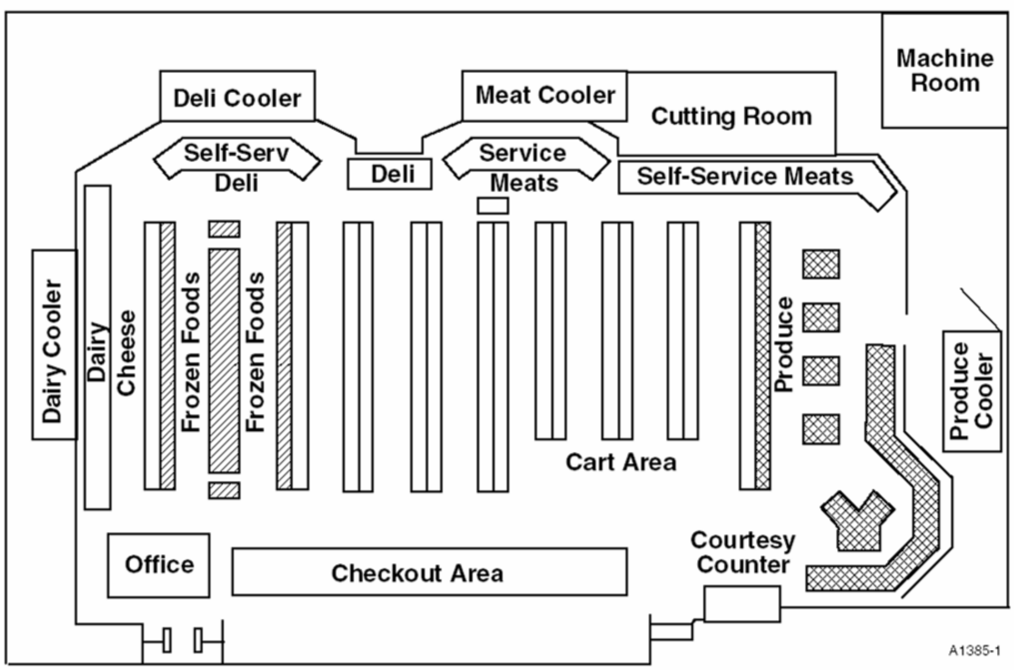

Figure 1. Layout of Refrigerated Display Cases and Walk-in Coolers in a Typical Supermarket

Figure 2 is a diagram of the most commonly used compressor arrangement in supermarkets, which is multiplex refrigeration. The term "multiplex" refers to the use of multiple compressors piped to common suction and discharge manifolds, all installed on a skid. The skid contains all necessary piping, control valves, and electrical wiring to control the compressors and the refrigeration provided to the display cases and walk-in coolers. The hot discharge gas from the compressors is piped to a remotely located condenser, which condenses the gas to a liquid. Liquid refrigerant returning from the condenser is piped back to the compressor rack, where a receiver, liquid manifold and associated control valves are located, distributing the liquid to the cases and coolers. Each case or cooler circuit is piped with liquid and suction return lines connected to the liquid and suction manifolds located on the compressor skid. Valves used for control of these circuits are located on the manifolds. The control valves employed consist of regulators to control suction pressure and solenoid valves used to control gas routing during defrost. The compressor rack is normally equipped with a number of pressure regulators used to control system head pressure, heat reclaim, and defrost. The rack will contain an oil separator in the discharge piping and an oil distribution system to return oil to the compressors.

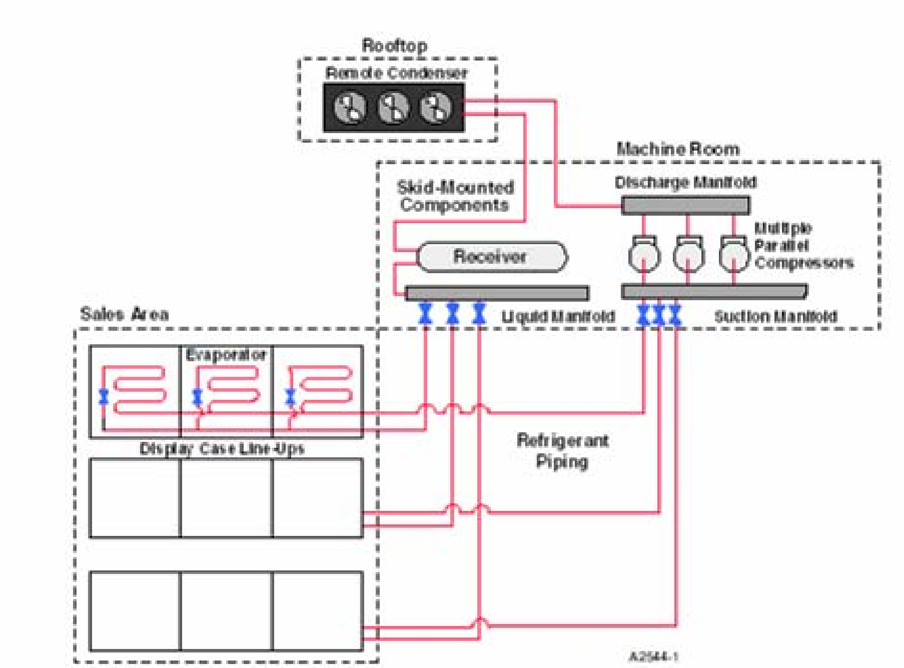

Figure 2. The Multiplex System is the Most Common Type of Refrigeration Used in Supermarkets

Typically, three to five compressor racks are employed to provide all refrigeration in the supermarket. The display cases and coolers are grouped, and attached to the compressor racks based on required saturated suction temperature (SST) to maintain the desired case temperature. A supermarket will have 1 or 2 low temperature racks to address all frozen food refrigeration requirements. The low temperature racks will typically operate at a –20°F SST. Refrigeration loads as low as –30°F, or as high as –10°F, will also be provided by the low temperature racks. In these situations, the suction manifold will be divided, and one or two compressors will provide the off-temperature refrigeration. The discharge of these "satellite" compressors will be piped to the common discharge manifold with the other low temperature loads so that a common condenser and liquid manifold can be used for all circuits on the rack. The remaining refrigeration circuits in the store are referred to as medium temperature and normally require a 20°F SST. Two or more compressor racks are needed to meet all medium temperature refrigeration requirements. Satellite compressors are also used for medium temperature loads requiring an SST significantly higher or lower than 20°F.

Multiplex systems commonly consist of three or four compressors that are sized such that operation of all compressors simultaneously can provide adequate capacity to meet the design refrigeration load. During off-design operation, the refrigeration load can be considerably less than the design value; at the same time, the refrigeration capacity of the compressors can increase with a decrease in ambient temperature and the condenser operating pressure. In this situation, the compressors can be selected so that the combined capacities of the compressors closely match the refrigeration load. The selection of, and the on-off cycling of the compressors is done based on suction pressure value measured at the compressor rack. The use of microprocessor-based controls allows more sophisticated control algorithms to be employed, so that very close matching of the suction pressure and the set point value can be maintained with multiplex compressor systems.

The most common type of condenser used in supermarket refrigeration is air-cooled. The reason is air-cooled condensers require the least maintenance and have been shown to operate reliably in the non-operator environment of supermarket refrigeration. Air-cooled condensers employ finned coil construction with 8-10 fins/inch and multiple fans. On/off fan cycling is used as a means to control condensing temperature and to reduce fan energy at ambient temperatures below design.

Evaporative condensers are used in some supermarkets, primarily in states like California with a drier climate, where a substantial difference in dry-bulb and wet-bulb temperatures exists. The evaporative condenser consists of a tube bundle, a fan for air flow, and a water sump and pump system used to spray water over the tube bundle. The refrigerant vapor is passed through the tube bundle where heat is removed and the refrigerant is condensed. The resulting condensing temperature can be close to the ambient wet-bulb. Evaporative condensers require less air flow than air-cooled condensers of equivalent rejection capability and can, therefore, be operated at a lower minimum condenser temperature without a fan energy penalty.

Water treatment and consumption are major issues that prevent more use of evaporative condensers in supermarkets. Water treatment is needed because of the evaporation of the water, which concentrates dissolved minerals. The minerals will precipitate and form deposits on tube surfaces. Exposure of the water to air causes biological growth within the evaporative condensers in the form of algae and slime. Treatment of evaporative condenser water consists primarily of "blow down" in which a fraction of the water is discharged to drain and replaced with fresh water. The amount of blow down is controlled by a conductivity controller. The discharged water will carry away excess minerals and solids, which prevents precipitation. Biocides, such as chlorine, are added to the water by automatic feeders to prevent biological growth.

The use of evaporative condensers is strongly recommended when energy-efficiency is an issue, because of the lower average condensing temperatures achieved with this type of heat rejection.

1.2 Low-charge Refrigeration System Options

As a result of using the layout described above in supermarkets, the amount of refrigerant needed to charge a supermarket refrigeration system is very large. A typical store will require 3,000 to 5,000 lb of refrigerant. The amount of piping and pipe joints present in supermarket refrigeration systems leads to significant refrigerant leakage. This leakage can amount to a loss of up to 30 to 50% of the total charge annually (1).

With increased concern about the impact of refrigerant leakage on global warming, new supermarket system configurations requiring significantly less refrigerant charge are now being considered. Advanced systems of this type include:

- Low-charge multiplex - Several refrigeration system manufacturers now offer control systems for condensers that limit the amount of refrigerant charge needed for the operation of multiplex refrigeration resulting in reducing the charge by approximately 1/3.
- Distributed – Multiple scroll compressors are located in cabinets placed on or near the sales floor. Scroll compressors are employed to minimize system noise in the sales area. The cabinets are close-coupled to the display cases and heat rejection from the cabinets can be done through the use of a glycol loop that connects the cabinets to a fluid cooler, in order to minimize refrigerant charge.
- Advanced self-contained – Self-contained refrigeration consists of compressors and condensers built into the display cases. An advanced version of this concept would use horizontal scroll compressors with capacity control, such as unloading, and water-cooled condensers with a water loop for heat rejection. The advanced self-contained refrigeration system would employ the smallest refrigerant charge.
- Secondary loop – A secondary fluid loop is run between the display cases and a central chiller system. The secondary fluid is refrigerated at the chiller and is then circulated through coils in the display cases where it is used to chill the air in the case.

While all of these systems use less refrigerant, energy use varies and can be greater than the energy usage with centralized multiplex racks. This is particularly true for the advanced self-contained system where only small, inefficient compressors are available for use in individual display cases. Both the distributed and self-contained systems involve the use of compressors and other mechanical hardware on or near the sales area, which is an arrangement not favored by supermarket operators.

In contrast, the secondary loop refrigeration system employs central mechanical systems and reduces the total refrigerant charge to approximately 300 to 500 lb. The anticipated refrigerant leak rate for the secondary loop refrigeration system is much less than that of the direct-expansion systems, because all refrigerant is contained within the chiller system.

Another significant benefit derived from the operation of secondary loop refrigeration is improved product temperature of stored perishables. Information provided in (2) suggests that with properly designed coils, display cases using secondary fluid for refrigeration produce a more uniform air temperature throughout the case than seen in direct-expansion display cases. When warm brine is employed for display case defrost, the time needed for defrost is significantly shorter, which prevents warming of the product. The resulting product temperatures are lower and more consistent than the product temperature recorded in direct expansion cases operating at the same air temperature.

Secondary loop refrigeration has been used extensively in Europe, particularly for medium temperature applications (3). These present systems, in general, have used more energy than comparable multiplex refrigeration. The reasons for the increased energy use include:

- The use of display cases with evaporators designed for direct-expansion operation – heat transfer characteristics of these coils are not suited for the use of the secondary fluid. The coil must be operated at a lower temperature to achieve the desired refrigeration load and case temperature.
- Operation of the refrigeration system at lower saturated suction temperature – caused by the added temperature difference of the secondary fluid that must be provided to maintain the required refrigeration temperature. The lower saturated suction temperature reduces the efficiency of the refrigeration compressors, increasing refrigeration energy use.
- Pumping energy associated with the circulation of the secondary fluid. Many of the secondary loop systems employ propylene glycol and water as the secondary fluid, because it is non-toxic and acceptable for use near foods. At low temperature, the viscosity of the glycol-water mixture is much higher than that of water alone, greatly increasing the fluid friction and pumping power.

Presently, the installed cost of the secondary loop refrigeration system may be higher than that of multiplex refrigeration. The reason for this increased cost is the multiplex system is established and there are many qualified contractors, which leads to cost competition. It is very likely that increasing the number of installations of secondary loop refrigeration could also cause the first cost of these systems to drop.

Supermarket operators can justify the higher first cost of the secondary loop system, if the benefits derived from its use provide an acceptable payback and return on investment. The environmental and food storage benefits help offset the added cost, but the added operating cost of the energy penalty reduces the payback potential.

The purpose of this project is to address the energy consumption of a secondary loop refrigeration system and find ways to reduce it. Since the secondary loop system employs a chiller system similar to the multiplex system, many of the energy-saving features now used with multiplex systems, such as floating head pressure, mechanical subcooling, and evaporative heat rejection can be implemented to reduce energy use. Certain unique characteristics of the secondary loop system can also be exploited for energy savings. These characteristics include: close-coupling between the chiller evaporator and the compressor suction which reduces pressure drop and refrigerant gas superheating; and liquid refrigerant subcooling by brine warming for defrost. The use of a low viscosity fluid and the secondary fluid along with better pump capacity control can greatly reduce the energy associated with brine pump. Finally, the use of display cases and fixtures with heat transfer surfaces designed specifically for secondary loop systems can greatly reduce heat transfer penalties.

Initial estimates for an advanced secondary loop system employing all of these design and energy-saving features suggest that the annual energy use could be significantly less than that of the multiplex refrigeration system now employed in many supermarkets.

1.3 Project Objectives

The overall goal of this project was to determine if a high-efficiency secondary loop refrigeration system could be economically attractive, due to reduced operating costs, than a state-of-the-art multiplex refrigeration system, which contains significantly more refrigerant.

This project contributes to the PIER program objective of reducing the environmental costs of California’s electrical system, by developing an alternative refrigeration system, which uses significantly less refrigerant than conventional refrigeration systems. It also contributes to the PIER program objective of improving energy cost/value of California’s electricity by lowering electrical consumption of supermarket secondary loop refrigeration systems.

The project identified system improvements to achieve maximum energy efficiency for secondary loop refrigeration systems in supermarkets.

The specific technical objectives of the project were to design and test a secondary loop refrigeration system, which:

- Consumes approximately 14% less electricity than a state-of-the-art multiplex refrigeration system (baseline system) installed in a comparable store.
- Has a refrigerant charge that is one tenth of (less than 500 lbs.) the baseline system.
- Loses annually no more than 15% of its refrigerant charge due to leakage.

The specific economic objective of the project was to determine if the prototype, high-efficiency secondary loop refrigeration system has lower total operating costs (operations, maintenance and repair) than the baseline system.

1.4 Project Organization and Approach

Southern California Edison RTTC and its subcontractor, Foster-Miller, Inc. obtained support from Safeway, Inc. to provide a supermarket where an advanced secondary loop refrigeration system could be installed and tested. After design and analysis work was performed, a specification for the advanced secondary loop system was prepared and several refrigeration manufacturers were asked to propose and bid on the construction. Hill PHOENIX, Inc. of Conyers, GA was selected as the system manufacturer and installing contractor based on their response to the request for proposal.

The advanced secondary loop refrigeration system was installed and instrumented for evaluation of its performance. Concurrently, a second supermarket operated by Safeway was identified in which a state-of-the-art multiplex refrigeration system was installed. The multiplex system was instrumented for performance measurement and both sites were monitored for approximately 9 months. Originally, the test plan called for a one-year field test, but problems with the data acquisition system at the multiplex store forced the testing to be truncated.

The measurements were recorded from both sites and compared with respect to the following variables:

- Refrigeration electric energy consumption
- Power demand – The peak power demand for the refrigeration at each site was determined and the impact on the total store demand was developed.
- Energy Efficiency Ratio (EER) –defined as:

$$EER = \frac{\text{Refrigeration Supplied}}{\text{Power Required}}$$

- Product temperature – Product sensors in similar cases were provided at each site so the average product temperature and temperature uniformity could be compared

Refrigerant replenishment data for each site were obtained from Safeway so the leak rates and annual refrigerant cost could be found and compared.

1.5 Report Organization

This report is organized as follows:

Section 1.0	Introduction
Section 2.0	Project Approach
Section3.0	Project Outcomes
Section 4.0	Conclusions and Recommendations
Section 5.0	References

2.0 Project Approach

2.1 Design of the Secondary Loop Refrigeration System

2.1.1 Description of the Secondary Loop Refrigeration System

Figure 3 shows the elements of the secondary loop refrigeration system. The difference between secondary loop and direct expansion refrigeration in supermarket applications is refrigeration of the display cases and walk-in coolers is provided by a chilled, secondary fluid, which is pumped between the refrigeration system and the refrigerated fixtures. In direct expansion systems, liquid refrigerant is piped directly to each fixture where it is flashed and evaporated to produce cooling. The resulting refrigerant vapor is piped back to the central compressor system, completing the refrigeration cycle. Secondary loop systems have all of the refrigeration associated with liquid chilling located within the central machine room of the supermarket. The chiller system resembles the multiplex refrigeration system since multiple parallel compressors are employed in both systems. The compressors are utilized based upon suction pressure which controls the supply temperature of the chilled fluid. The use of multiple compressors allows the refrigeration capacity to conform to changing operating conditions, resulting in better fluid temperature control and lower compressor energy use. The cooling of the secondary fluid takes place in a heat exchanger whereby the liquid refrigerant of the chiller system is evaporated to provide chilling to the secondary fluid. Heat rejection for the chiller system is accomplished utilizing a condenser cooled by ambient air. Either a dry, air-cooled condenser or an evaporative condenser may be employed. The evaporative condenser is favored because of the lower condensing temperature which may be obtained by rejecting heat to the lower ambient wet-bulb temperature. The lower condensing temperature provides greater compressor refrigeration capacity and lower power draw, both of which result in reduced refrigeration energy consumption.

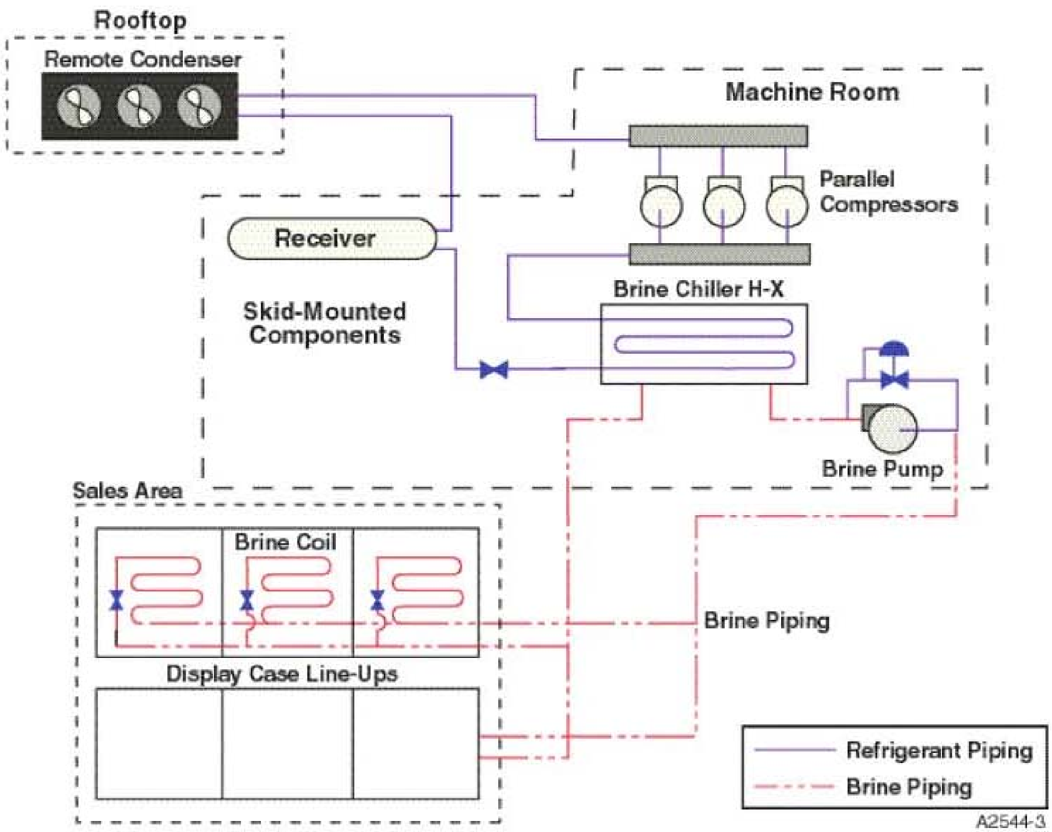

Figure 3. Flow Diagram for a Secondary Loop Refrigeration System

The secondary loop refrigeration can be configured to operate with two to four separate secondary fluid loops and chiller systems. In the two loop configuration, the secondary fluid temperatures are 20 and -20°F for medium and low temperature refrigeration. As a consequence of using only two temperatures for refrigeration, all display cases and storage coolers must operate with these two temperatures. For this and other reasons, the heat exchangers in the cases and coolers must be sized differently than those used for direct expansion operation. Additional loops may be employed if a secondary fluid temperature addresses a large fraction of the refrigeration load and this temperature is significantly different from 20 or –20°F. Possible examples of alternate loop temperatures are values of –10, 0, or 15°F, depending upon the air temperature required by these refrigeration loads. The use of multiple secondary fluid loops with temperatures closely matching the case air temperature requirements can result in more energy efficient operation of the secondary loop system.

Secondary fluid loop piping consists of which convey all fluid flow to and from the chiller. The piping terminates with branches which feed chilled fluid to the display case lineups. The piping for the main and branches is typically sized at a fluid velocity of 4 – 6 ft/sec. In each lineup, further branching occurs to provide flow to each display case coil. A temperature control valve is used for each case coil to control case air temperature. The fluid flow is regulated to maintain the air temperature set point. The branch piping is equipped with balancing valves, used to set the branch flow rate. This adjustment is normally done at installation.

The piping for the secondary fluid loops may be steel, copper, or plastic. The recommended type of plastic piping is constructed of high-density polyethylene (4). Insulation is also required. For the medium temperature piping, the recommended insulation is closed-cell foam (4). The recommended thickness of the insulation is ½ - ¾ in. for piping underground or in air-conditioned space, 1 ½ in. for pipe in non-conditioned space. For the low temperature piping, the insulation should be either Styrofoam or Polyisocyanurate foam. The recommended insulation thickness for low temperature piping is 1 in for piping in air-conditioned space and 1 ½ to 3 in. in non-conditioned space.

Secondary fluid flow rates are moderately high, since it is desirable to limit the temperature change of the fluid to 7 to 10°F while refrigerating the display cases. The resulting total flow rate for each loop is as high as 300 to 500 gpm. Because of the high viscosity of the secondary fluid at refrigerating temperatures, and the fact the fluid is circulated continuously, the energy associated with pumping is substantial and therefore, is a major component of the overall energy consumption of the secondary loop system.

The central chiller systems employ multiple compressors that are piped in parallel. The compressors are multiplexed and are on/off cycled in response to the suction pressure of the chiller evaporator.

Because of the location of the evaporator on the chiller skid, the compressors for the secondary loop system are considered close-coupled to the evaporator. The pressure drops and return gas heat gain are minimized in this configuration. Both these factors help reduce compressor energy consumption.

The chiller system for the low temperature refrigeration may be equipped with mechanical subcooling. A direct expansion heat exchanger is used to subcool the liquid refrigerant used by the low temperature chiller system.

Secondary loop refrigeration systems employ warm secondary fluid to defrost the display cases. In this system, a heat exchanger at the chiller is used to heat a portion of the fluid using discharge gas from the compressors. The warm fluid is piped to the cases using an additional pipe. The fluid leaving the defrosted case is returned to the chiller through the common return piping system. The use of warm fluid to defrost has been found to shorten defrost time substantially (2) and also reduce the amount of refrigeration associated with pull-down and recovery of the case back to operating temperature. It is estimated that the energy use for defrost is half that used by multiplex systems using hot gas defrost (4).

2.1.2 High-Efficiency Features for Secondary Loop Refrigeration

The purpose of the research project is to investigate methods which maximize the energy efficiency of the secondary loop refrigeration system. The features chosen for this purpose include the following:

- Display cases designed for use with secondary fluid
- High-efficiency reciprocating compressors
- Close-coupling of the evaporator to the compressor system
- Multiple-parallel pumps
- Evaporative heat rejection
- Low viscosity secondary fluid
- Warm-fluid defrost

2.1.3 Display Cases Designed for Use with Secondary Fluid

A significant factor affecting the energy consumption of the secondary loop system is the SST at which the refrigeration must operate to meet the temperature requirements of the display cases. Without proper display case design, the SST of the secondary loop system can be lower than that seen in a comparable direct expansion refrigeration system because of the added temperature difference associated with the secondary fluid. In previously installed secondary loop systems, the display cases were often the same as those used in direct expansion systems. The heat exchanger coils in these cases were designed specifically for refrigerant and evaporation and did not provide sufficient heat transfer surface for secondary fluid use. The heat exchangers used required a larger temperature difference between the secondary fluid and the case air to achieve the desired display case air temperature.

Prior to this project, Hill PHOENIX, Inc. (manufacturer and contractor for the field test secondary loop system) had initiated an extensive redesign of the display case heat exchanger coils. As a result the coils are better suited for use with secondary fluid. The heat exchanger coils are larger than those used for direct refrigerant expansion, resulting in a lower temperature difference to achieve the desired air temperature.

Table 1 shows a comparison of the required operating conditions for secondary loop and direct expansion refrigeration for similar display cases. For each display case type, the discharge air temperature of the case is listed. The expected saturated evaporator temperature and compressor rack SST needed to maintain this air temperature with direct expansion refrigeration is shown. The comparable secondary fluid supply temperature and SST of the chiller are also provided. For the cases listed, the SST values of the compressor rack and the chiller are the same, indicating the secondary loop system has no energy penalty because of the use of the fluid loop. There are two reasons for this result: the first is that the heat exchangers of the secondary loop display cases have been sized to utilize a fluid at approximately the same temperature as the direct-expansion display cases; and secondly, the compressors of the direct expansion system are located remotely from the case evaporators and a pressure drop occurs in the refrigerant suction lines which forces the compressors to operate at an SST that is significantly lower than the saturated evaporator temperature at the display case. The secondary loop refrigeration system has the evaporator chiller and compressors close coupled (explained in detail below), which eliminates the pressure drop due to the piping runs necessary for the direct expansion system.

Table 1. Comparison of Operating Temperatures for Direct-Expansion and Secondary-Fluid Refrigerated Display Cases

Display Case & Discharge Air Temperature		Direct-Expansion Refrigeration		Secondary-Fluid Refrigeration	
Display Case	Discharge Air Temp (°F)	Evaporator Sat. Temp (°F)	Compressor Rack SST (°F)	Secondary Fluid Supply Temp (°F)	Chiller SST (°F)
Frozen Food Reach-in	-12	-19	-25	-20	-25
Single-deck Meat	28	21	15	20	15
Multi-deck Deli	29	21	15	20	15
Multi-deck Produce	32	24	20	25	20

2.1.4 High-efficiency Refrigeration Compressors

The chiller system may employ either reciprocating or screw compressors. Screw compressors can provide the large refrigeration capacities needed to address the refrigeration load associated with each fluid loop. Screw compressors do not exhibit as high an energy efficiency value as reciprocating units, but are less susceptible to liquid refrigerant damage and, in general, are considered to be less of a maintenance issue. Reciprocating compressors may be used in the chiller system for better energy use characteristics.

Both reciprocating and screw compressors were evaluated for the advanced secondary loop refrigeration system. High-efficiency reciprocating compressors were chosen, because of the potential for energy savings.

2.1.5 Close-Coupling of Compressors and Evaporator

One of the energy-saving features of the secondary loop system is the close-coupling of the compressor system to the evaporator of the chiller. With this arrangement, the pressure drop between the evaporator and the compressors and the heat gain to the suction gas are minimized. Direct expansion systems incur substantial energy penalties because of the long pipe runs between the display cases and compressor racks, which result in pressure drops of 2-3 psi and as much as 40°F suction gas heating. Both of these penalties reduce the mass flow rate of the compressors, which results in loss of refrigeration capacity. Compressor run times are increased in direct expansion systems, which increase compressor energy consumption.

The evaporator chillers were sized so the difference between the chiller evaporator and the brine supply temperatures was minimized to a delta T of 5°F. This small difference allowed the compressors to operate at a higher saturated suction temperature, which helps to maximize the EER.

2.1.6 Multiple Parallel Pumps

The design secondary fluid flow rates represent the maximum brine flow rate required to refrigerate the display cases. When the refrigeration load is at less than design, a flow control valve in each case throttles the brine flow to maintain discharge air temperature set point. Display cases in a defrost cycle do not require chilled brine flow. For the above reasons, the total brine flow of each loop is often considerably less than the design value. The total loop flow cannot be reduced if a single pump is employed. Throttling of the pump to reduce flow reduces total pumping head is required at the display cases which are in a cooling mode to insure proper brine flow through the heat exchanger. The standard approach is to provide a fluid by-pass at the pump so that the excess flow is returned to the suction of the pump, while the full pressure head is maintained. Pump power is constant when a bypass is used to reduce system capacity when the system requires a constant head pressure.

A variable-speed pump could be used, but the matching of the head-flow characteristics of the pump to the head and flow values needed in the brine loop can be difficult, if not impossible. Often, the pump must operate at full speed to meet the head required even though a lower flow rate is appropriate. The expense of a variable-speed drive for the flow rates associated with the secondary loop system for supermarkets is quite high and the energy savings often do not justify this added cost.

The approach investigated is the use of multiple, parallel pumps with each pump sized at approximately 1/3 of the design brine flow rate with the pump head chosen to meet the full flow condition. Pumps are added or subtracted based upon the pressure difference between the brine supply and return flow. A rise in the pressure difference indicates flow valves have restricted flow or closed, reducing the total brine flow. When a pressure rise occurs, a pressure switch stops one of the pumps. A continuing rise in pressure causes the second pump to stop. The third pump continues to run to insure that brine flow to the display cases is continuous. A lowering of the pressure difference signifies more brine flow is required by the system. One of the two remaining pumps is started and the second added pressure continues to drop to the start set point of the third pump.

2.1.7 Evaporative Heat Rejection

Two types of condensers are available for supermarket refrigeration systems, air-cooled or evaporative.

In dry climates, similar to the climate in California, the temperature difference between the dry-bulb and wet-bulb temperature is significant, and can be as high as 25°F. When an evaporative condenser is employed with a refrigeration system, the decrease in condensing temperature is substantial, resulting in significant energy savings in the operation of the refrigeration compressors (5). Ambient airflow required for an evaporative condenser is less than a comparably sized air-cooled unit; therefore, the fan power of an evaporative condenser is also less.

The use of evaporative condensing for the secondary loop system is recommended to minimize energy use.

2.1.8 Low Viscosity Secondary Fluid

The pumping power needed for the secondary fluid loops is greatly influenced by the thermal capacitance (the product of the fluid density and specific heat) and viscosity of the fluid. Higher thermal capacitance means that less mass flow is required to meet a refrigeration load, reducing the pumping power. Viscosity is a key parameter of fluid friction, and greatly influences pumping power. This is of particular importance for low temperature refrigeration applications, since fluid viscosity increases as temperature decreases. Secondary loop systems often employ propylene glycol-water solutions as the brine, wherein the concentration of the glycol is set to avoid ice formation. Glycols are preferred because they are inert to all common piping materials and most non-metallic gaskets and seals. Propylene glycol is also non-toxic and non-flammable and is the most suitable glycol for use where food products are involved. A 30% propylene glycol / water mixture shows reasonable heat transfer and pumping properties for medium temperature (+20°F fluid temperature). In low temperature refrigeration, the brine temperature will fall in the range of -20 to -30°F. The required concentration of glycol for anti-freeze in this temperature range is 50% and the resulting viscosity is 156.1 centipoise (for comparison, the viscosity of water at 68°F is 1.0 centipoise).

Alternative secondary fluids to glycol-water solutions exist that will remain liquid at the required operating temperatures and also have better transport properties. These fluids have been evaluated by others (6). The results of these evaluations show the preferred secondary fluid is an organic-salt and water mixture, which has lower viscosity than propylene glycol-water and also has higher thermal capacitance.

The organic-salt and water solution investigated in this project is marketed under the trade name of Dynalene. Table 2 gives a comparison of the relevant transport properties of propylene glycol-water and Dynalene at fluid temperatures of -20 and 20°F, which are standard values for low and medium temperature refrigeration applications. Two Dynalene compositions specified are HC-30 and HC-10 for low and medium temperature refrigeration, respectively. The differences in properties between propylene glycol-water and Dynalene are most notable at -20°F, particularly in terms of viscosity.

Table 2. Transport Properties of Propylene Glycol-Water and Organic Salt-Water (Dynalene) at Operating Temperatures of Secondary Loop Refrigeration

Property	Propylene Glycol- Water	Dynalene
Brine temperature = + 20°F		
Concentration (% by weight)	30	HC-10
Density - ρ (lb/ft^3)	64.9	75.0
Specific Heat - C (Btu/lb- °F)	0.90	0.80
Thermal Capacitance (ρ×C)	58.4	60.0
Viscosity (centipoises)	9.91	2.84
Brine Temperature = - 20 °F		
Concentration	50	HC-30
Density - ρ (lb/ft^3)	66.5	81.2
Specific Heat - C (Btu/lb- °F)	0.8	0.77
Thermal Capacitance (ρ×C)	53.2	62.5
Viscosity (centipoises)	156.1	8.78

The impact of the fluid transport properties on the energy consumption of secondary loop refrigeration can be determined through evaluation of the relative pumping power (4,6).

The relative pumping power is found by holding the refrigeration load, fluid temperature difference, pipe diameter and pipe length constant. The relationship of heat transport, fluid friction and pump power are combined to give an expression for pumping power in terms of fluid properties only, relative to a base fluid:

$$\frac{\text{Pumping Power}_\text{A}}{\text{Pumping Power}_\text{B}} = \left(\frac{\rho_\text{A}}{\rho_\text{B}}\right)^{-1.8} \left(\frac{\mu_A}{\mu_B}\right)^{0.2} \left(\frac{\text{c}_\text{A}}{\text{c}_\text{B}}\right)^{-2.8}$$

Using propylene glycol-water as the base fluid (B), the relative pumping power of Dynalene are 0.44 and 0.84 for fluid temperatures of -20 and 20°F, respectively.

These results indicate the use of lower viscosity fluid in both the medium and low temperature fluid loops will reduce energy use for brine pumping. For the advanced secondary loop refrigeration system, Dynalene was specified for all fluid loops.

2.1.9 Refrigerant Subcooling from Warm Brine Defrost

During operation, the cooling coils of the refrigerated display cases and walk-in coolers develop frost build-up from condensation and subsequent freezing of water vapor from air passing through the coils. The frost must be removed periodically. Otherwise the coils will clog, preventing airflow and reducing the cooling capability of the fixture. The secondary loop refrigeration system employs a flow of warmed brine for this defrosting process. Brine is taken from the refrigeration loop and heated with refrigerant rejection heat to a temperature of approximately 65 or 80°F for medium and low temperature refrigeration, respectively. The warm brine flows to the display cases by a separate piping loop used exclusively for defrosting. When a display case is scheduled to defrost, a valve in the warm brine supply line opens, providing warm brine to the display case. The brine passes through the case heat exchanger where heat from the brine melts the frost. The brine then flows to the return piping of the loop where it is pumped back to the chiller system. It is important to note that the melting of the frost chills the brine close to the normal return temperature therefore little excess heat is added to the total refrigeration load by the defrosting.

The heating of the brine for defrost is done with two heat exchangers at the chiller system. The first exchanger uses heat from the liquid refrigerant coming from the system receiver. The second heat exchanger uses the heat contained in the compressor discharge gas to raise the temperature of the brine to the desired temperature.

The initial heating of the brine by the hot liquid refrigerant results in lowering significantly the temperature of the refrigerant, producing a subcooled liquid. The use of this sub-cooled refrigerant liquid in the chiller system increases the chilling capacity of the system, which means fewer compressors need to operate to satisfy the refrigeration load. The reduction in compressor run time results in a reduction in compressor energy used.

Field testing showed the refrigerant subcooling produced by the warm brine defrost averaged about 12°F, which corresponds to capacity increases of 6.9 and 6.4% for low and medium temperature refrigeration, respectively.

2.2 Analysis of the Secondary Loop and Multiplex Refrigeration Systems

Two refrigeration system models were formulated for the analysis work done on this project. The first model was used to determine the energy consumption of a direct expansion multiplex refrigeration. The second model for the secondary loop system used a modified version of the multiplex system to describe the compressors and condensers. Models for the brine loops and pumping were then added to the compressor and condenser models to form the full secondary loop refrigeration system.

2.2.1 Multiplex Refrigeration Model

The annual performance of the multiplex refrigeration system is calculated on the basis of ambient dry-bulb temperature bins, where each bin specifies: an ambient dry-bulb temperature value; the coincident value of the wet-bulb temperature; and the number of hours at which the ambient temperature occurs during the year. The general procedure is, for each temperature bin, to calculate the power needed by the refrigeration system and apply that power to the number of hours at each ambient temperature. The procedure is repeated for all ambient temperatures present at the site.

A refrigeration configuration must also be specified, where each refrigeration system employed in the supermarket is described in detail. System information identifies the design refrigeration load, display case evaporator temperature in the system, minimum condenser temperature, type of condenser (air-cooled or evaporative), and the refrigerant employed. Other information specified includes saturated temperature change between the display cases and the suction of the compressors, which is representative of the suction pressure drop, and the temperature rise in the return gas.

The first step in the analysis is to determine the refrigeration load on the system. Past experience (7) has shown that the refrigeration load will vary with outside ambient dry-bulb temperature, decreasing as the ambient temperature decreases. The rate of decrease in load is greater for medium temperature than low temperature refrigeration, and a minimum is reached, which is a result of space heating the store. Store temperature will not fall below 68°F. The minimum store temperature is normally present at outside ambient temperatures less than 40°F. The maximum values of store temperature and humidity are also constrained by the use of air conditioning, which tends to maintain the temperature at 75°F with a corresponding relative humidity of 55%. Maintaining the indoor conditions at these levels can be expected at outdoor ambient temperatures of greater than 85°F. Based on these constraints, a load factor can be calculated which is applied to the design load to determine the refrigeration load for each temperature bin. The refrigeration load factor is determined from the following relationships:

For both low and medium temperature refrigeration, the value of the refrigeration load is set at design for ambient temperatures of 85°F or greater.

At ambient temperatures of 40°F or less, the refrigeration load is at its smallest fraction of design value, which is 66% for medium temperature and 80% for low temperature.

For ambient temperatures between 40 and 85°F, the load factor is found from:

$$\text{Load factor } = \left(1 - \left(1 - \min\right) \frac{(85 - T_{amb})}{(85 - 40)} \right)$$

where

> min = the minimum fraction of design load (0.66 for medium temperature and 0.8 for low temperature
> T_{amb} = ambient dry-bulb temperature

The state points for the multiplex refrigeration system are then determined. The system configuration specifies the desired evaporator temperature for the display cases. In operation, pressure drop will occur between the case evaporators and the compressor suction. This drop is reflected in a lower saturated temperature value at the compressor suction. Heat gain to the return gas will also take place, which affects the mass flow rate of refrigerant seen by the compressor. Both of these factors tend to decrease the capacity of the compressors and increase the run time need to meet the refrigeration load. The amount of pressure drop and superheating is a strong function of the distance between the display cases and the compressors, increasing with increased distance. In the analysis, these factors are taken into account by values included with the system configuration description.

The condenser temperature is also determined at this time. The most significant parameter in determining condensing temperature is the outdoor ambient temperature, since heat is rejected to ambient conditions. The operation of the condenser can be characterized by the temperature difference (TD) between the condensing refrigerant and the ambient. The condensing temperature is allowed to vary with the ambient until a certain minimum condensing temperature is reached. At that point, control of the condenser fans or a liquid pressure regulator maintains the condensing temperature at the minimum value. The model compares the ambient temperature with the characteristic TD of the condenser type specified and calculates a condensing temperature. The calculated value is compared with the specified minimum condensing temperature. If the calculated temperature is less than the minimum, the minimum value is used to set the state point of the refrigeration system.

The condenser TD is determined by the type of condenser modeled. For air-cooled condensers, the TD refers to the difference between the condensing and ambient dry-bulb temperatures. The standard values of TD for air-cooled condensers in supermarket refrigeration are 10 and 15°F for low and medium temperature, respectively. The TD of an evaporative condenser refers to the difference between condensing and ambient wet-bulb temperatures. Evaporative condensers are often sized to produce a condensing temperature of 100°F at the design wet-bulb, however, analysis and field measurements by Foster-Miller (8) showed the lowest combined compressor and condenser fan power is achieved at a TD value of approximately 12°F. This lower value was used for all relevant analysis.

The refrigerant liquid temperature is determined as part of the state points of operation. For non-subcooled systems, the liquid temperature is 10 degrees less than the condensing temperature. When mechanical subcooling is employed in multiplex systems, the liquid refrigerant temperature leaving the subcooler heat exchanger is typically 40°F. Some warming of the liquid occurs as the liquid is piped to the display cases .The temperature of the liquid entering the cases is set at 42°F. The refrigeration needed for mechanical subcooling is normally provided by the highest temperature refrigeration system in operation, typically at a SST of 25 - 35°F. The size of the mechanical subcooling load varies with the load of the subcooled system, normally the low temperature refrigeration. For each temperature interval the low temperature refrigeration load is first determined, and the liquid refrigerant flow required for this load is then determined. The subcooling load is the amount of cooling needed to lower the temperature of the liquid flow from 10°F less than the condensing temperature to 40°F. The subcooling load is added to the refrigeration load of the medium temperature system and is used to determine the energy consumption for that system.

Once the state points are determined, the capacity and power of the compressors is found. These calculations are made using the compressor performance data supplied by the manufacturers. Performance data consists of refrigeration capacities (Btuh), refrigerant mass flow rate (lb/hr), and input power (Watts) as functions of saturated suction temperature (SST) and saturated discharge temperature (SDT). These data are obtained from the ARI equation:

$$\begin{aligned}\text{Capacity, Mass Flow, or Power} = {} & C_0 + C_1{*}SST + C_2{*}SDT + C_3{*}SST^2 \\ & + C_4{*}SST{*}SDT + C_5{*}SDT^2 + C_6{*}SST^3 \\ & + C_7{*}SDT{*}SST^2 + C_8{*}SST{*}SDT^2 \\ & + C_9{*}SDT^3\end{aligned}$$

where

$C_0 \ldots C_9$ = Performance equation coefficients
SST = Saturated suction temperature (°F)
SDT = Saturated discharge temperature (°F)

Compressor manufacturers provide three sets of coefficients for each compressor, where each set is to determine either cooling capacity, mass flow rate, or compressor input power.

The compressor cooling capacity and mass flow rate given by the above equations are determined at particular rating conditions. One such condition commonly seen is a return gas temperature of 65°F and 0°F of liquid subcooling. Corrections are made to account for the values of superheat and refrigerant liquid temperature. The superheat correction takes into account the density and enthalpy change, while change in liquid temperature affects the enthalpy of the refrigerant entering the evaporator. The correction applied to the compressor capacity or mass flow rate is:

$$Capacity\ \text{correction} = \frac{\mathrm{v_r}}{\mathrm{v}} * \frac{(\mathrm{h_{out}} - h_{in})}{(h_{rout} - h_{rin})}$$

where

v = the specific volume of the refrigerant (ft^3/lb)
h_{in} = the enthalpy of the refrigerant entering the evaporator
h_{out} = the enthalpy of the refrigerant leaving the evaporator
the subscript, r, designates that the property is at the rating conditions

The capacity value and refrigeration load are then used to find the number of compressors operating by taking the ratio of refrigeration load to capacity. Typically, three or four compressors are needed to meet the load at design conditions. At other conditions less than this number is required. Fractional values represent compressor on/off cycling. The analysis does not use specific compressor models, but instead uses a single generic size for each type of compressor. The generic size is based upon the most commonly compressor, which is a 7.5 HP unit for the reciprocating compressor and a 6 HP unit for a scroll compressor.

Compressor energy consumption for the temperature bin is found by first determining the power needed by the compressor at the state point and load conditions. Compressor power is multiplied by the number of compressors operating and the number of hours associated with the ambient temperature.

The fan power for remote condensers or fluid coolers is based upon the type of condenser or cooler being utilized. Air-cooled condensers for low temperature refrigeration are sized for a smaller TD and require more fan power than condensers employed with medium temperature refrigeration. Fan requirements are less for evaporative heat rejection than is needed for dry rejection, because less air flow is required. The power value listed in the table for the evaporative units includes the sump pump used to spray water over the heat exchanger coil.

The condenser fans operate continuously as long as the resulting condensing temperature is greater than the specified minimum value. Fan cycling is employed with both the condensers and fluid coolers to regulate the condensing temperature when full operation of the fans reduces condensing temperature below the minimum value. Fan energy is estimated by multiplying the installed fan power by a fan factor that represents the amount of fan operation needed to maintain the condensing temperature at the minimum. For air-cooled condensers and dry heat rejection, the analysis sets the fan factor at 1.0 when the sum of the ambient dry-bulb temperature and the condenser TD are greater than the minimum condensing temperature. The fan factor is set at 0.25 when the ambient dry-bulb temperature is less than 30°F. For ambient temperatures greater than 30°F where continuous fan operation is not needed, the fan factor is calculated from:

$$\text{Fan Factor} = \left(1 - 0.75 * \frac{(\mathrm{T_{con}} - \mathrm{TD} - \mathrm{T_{amb}})}{(\mathrm{T_{con}} - \mathrm{TD} - 30)}\right)$$

where

T_{con} = the minimum condensing temperature (°F)
TD = the temperature difference (°F)
T_{amb} = the ambient dry-bulb temperature (°F)

For evaporative rejection, the above relation is also employed, but the ambient wet-bulb is used instead of the dry-bulb temperature. The fan energy for the temperature bin is determined from the product of the installed fan power, the fan factor, and the number of hours in the bin.

The energy for mechanical subcooling of the low temperature refrigeration is addressed by the medium temperature system with the highest SST. As mentioned previously, the subcooling load is calculated for each temperature bin and is added to the refrigeration load of the appropriate medium temperature system. The compressor run time for the medium temperature system is calculated on the basis of this combined load.

The bin calculation is repeated until energy values are set for each temperature bin for the refrigeration system configuration specified. Once the bin loop is completed, the model then obtains the next system description and the bin loop is repeated for this next system. The procedure continues until all systems are analyzed.

2.2.2 Modeling of Secondary Loop Refrigeration

The major difference in the analysis of the secondary loop refrigeration system is the operation of the secondary loop. The loop consists of secondary fluid that is pumped between a central chiller and the display cases. Two to four fluid loops are employed in a supermarket, depending upon the composition of the refrigeration load. The analysis is, therefore, conducted separately for each of these loops at each ambient temperature. Energy results are combined at the completion of the analysis to determine total energy consumption.

The system configuration specifies the design refrigeration load for each secondary loop. The analysis first considers the variation on the refrigeration load seen at each ambient temperature, using the method described previously. The model assumes that the refrigeration load at the display cases is handled by a constant temperature change of the fluid, while the flow through the cases is varied as the refrigeration load varies. This flow arrangement is an attempt to simulate the operation of a temperature control valve that maintains constant fluid outlet temperature from the display case heat exchanger. Since all loads on the loop behave in this fashion, the estimated total fluid flow can be found from:

$$\dot{M}_{brine} = \frac{Q_{ref}}{C_{brine}\Delta T_{brine}}$$

where

Q_{ref} = the refrigeration load addressed by the brine loop (Btuh)

$\dot{M}_{brine}$ = the mass flow rate of brine (lb/hr)

ΔT_{brine} = the temperature change in the brine seen at the cases

C_{brine} = the specific heat of the brine

The secondary fluid loop will experience some heat gain, while flowing between the cases and the central chiller. The most significant of these gains is the addition of energy due to operation of the secondary fluid loop pump. The pump power is based on the maximum fluid flow needed to meet the design refrigeration load, which is found with the above relation. The required pump head is set at 75 ft of water column (WC) for low temperature refrigeration loop and 50 ft (WC) for the medium temperature loop. While the flow to the cases varies as the load changes, the total fluid flow through the pump remains constant, since a by-pass is used to regulate the operation of the pump in the loop. The power input to the pump is calculated as the ideal power for the maximum fluid flow and head divided by a pump efficiency of 55%. The power input to the pump is converted into heat in the fluid, which must be removed by the chiller system. The rise in temperature is calculated from:

$$\Delta T_{pump} = \frac{\text{Pump Power}}{\dot{\mathrm{M}}_{brine}\, C_{brine}}$$

where

Pump Power = the power input to the secondary fluid pump

Some line heat gain is also expected and was set at 0.25°F for the supply and return lines of the loop.

Once the total temperature rise of the secondary fluid loop is determined, the load on the chiller evaporator can be found by:

$$Q_{evap} = \dot{M}_{brine}\, C_{brine} \Delta T_{brine}$$

where

ΔT_{brine} = the total temperature gain seen in the fluid

Mechanical subcooling is used in the secondary loop system for the low temperature chiller system in the same fashion as is seen in multiplex systems. A portion of the medium temperature system provides the subcooling. The refrigeration load associated with the subcooling must be added to the total load of the medium temperature system. The subcooling load is calculated based upon the refrigeration load of the low temperature system, which sets the flow rate of refrigerant needed. The subcooling reduces the temperature of the refrigerant from the temperature leaving the condenser to 40°F. The sub-cooled liquid temperature is factored into the available capacity of the low temperature system in meeting its refrigeration load.

The state points of the chiller system are then determined. The evaporator temperature of the chiller heat exchanger is set at 7°F below the outlet temperature of the fluid loop.

The outlet temperature of the refrigerant is set at 8°F higher than the evaporator temperature. The temperature rise is due primarily to the control action of the thermostatic expansion valve of the chiller heat exchanger, which regulates the outlet temperature of the refrigerant at a superheated condition. Pressure drop of the refrigerant vapor to the suction of the compressors is negligible due to the close coupling of the compressors and the heat exchanger. The SDT of the compressor system is determined from the condensing temperature, which is calculated depending upon the type of heat rejection system analyzed. The secondary loop refrigeration system can be modeled with air-cooled, water-cooled, or evaporative condensing. The method of determining the condensing temperature for each heat rejection type is the same as described previously.

The central chiller uses multiple parallel compressors to address the chiller heat exchanger load. The types of compressors now employed in presently installed systems are either screw or reciprocating. Scroll compressors can be analyzed if desired. The procedure for determining the number of compressors operating is the same as that used for the multiplex and distributed systems. Manufacturer's performance data are used to determine compressor cooling capacity and power. The number of compressors operating is found from the ratio of the chiller cooling load to the total available compressor capacity. Compressor energy is the product of the compressor power, number of compressor operating, and the number of hours in the temperature bin being examined.

The energy consumption of the secondary fluid pump is determined using the method outlined previously. While the fluid flow to the display cases varies with changing refrigeration load, the total fluid flow across the pump is constant. Variation in flow is achieved by flow bypassing around the pump. The head addressed by the pump was estimated based upon the temperature of the secondary fluid loop, the secondary fluid employed, the length and diameter of pipe between the cases and chiller, and the pressure drop at the display cases and through the chiller heat exchanger. The secondary fluid examined here consists of an organic salt-water mixture (Dynalene) for the low and medium temperature loops. Loop piping diameter was sized to maintain a velocity of 4to 6 ft/sec, while the typical length of piping was estimated at 250 ft. Value of the pressure drop for the display cases ranges from 5 to 7 ft. (WC), while the pressure drop of the chiller heat exchanger was set at 20 ft (WC). The head requirements calculated for the secondary fluid loop pumps were 50 and 75 ft. (WC) for the medium and low temperature systems, respectively. The pump and motor efficiencies were taken at 55 and 85%, respectively.

During part-load operation, the flow rate of the secondary loops is less than design but the total head remains approximately the same due to the pressure drops through the operating display case coils. Pump system control was modeled as continuous. The model consisted of calculating the required fluid flow to meet the refrigeration load and then using that flow to calculate the pump power. The energy for that temperature bin was the product of the calculate pump power and the number of hours associated with that temperature bin.

The energy requirement for heat rejection is dependent upon the type of heat rejection employed. The energy is calculated using the procedures described previously.

2.3 Field Testing of the Secondary Loop Refrigeration System

2.3.1 Description of the Secondary Loop Refrigeration Test Site

The design and analysis work of the project were used to specify a secondary loop refrigeration system for an operating supermarket. A supermarket was chosen for the testing of the secondary loop system and is a store operated by Vons Supermarkets (a division of Safeway, Inc.). The supermarket is located in Thousand Oaks, CA and has a total building and sales areas of 48,351 and 41,912 ft^2, respectively. The layout of the store is shown in Figure 4 and the circuits that are refrigerated by the secondary loop system are listed in, Table 3, Table 4, and Table 5.

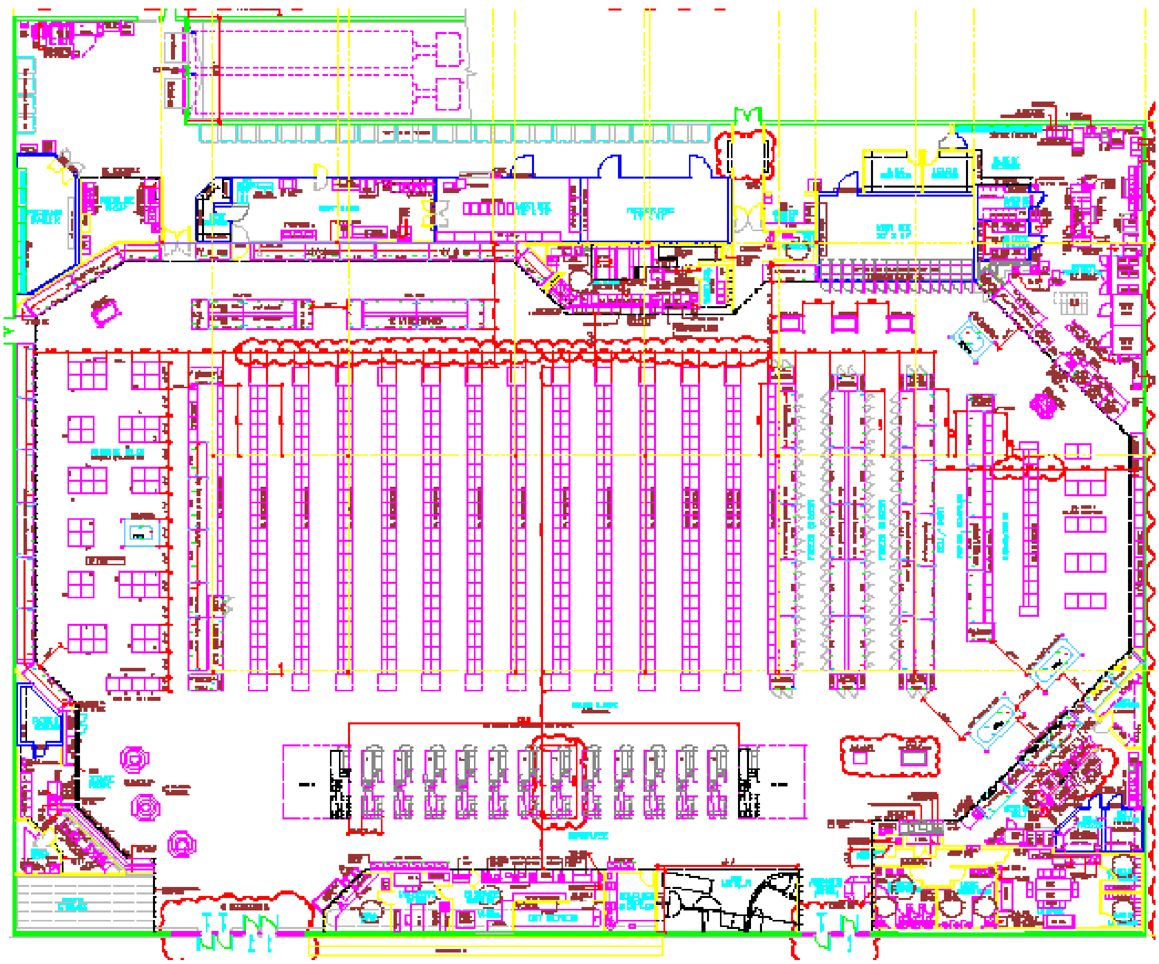

Figure 4. Layout of the Secondary Loop Refrigeration Test Store

Table 3. Low Temperature Circuits for the Secondary Loop Refrigeration Test Store (Loop A)

Circuit	Fixture Description	Refrig Load (Btuh)	Disch. Air Temp (°F)	Brine Flow (gpm)
A-1	Reach-in Frozen Pasta	3,000	-5	0.9
A-2	Wide Island Frozen Meat	19,800	-12	18.2
A-3	Wide Island Frozen Meat	17,400	-12	16.2
A-4	Freezer Walk-in Box	31,300	-12	24
A-5	Reach-in Frozen Food	24,000	-5	6.9
A-6	Reach-in Frozen Food	21,000	-5	6
A-7	Reach-in Frozen Food	24,000	-5	6.9
A-8	Reach-in Frozen Food	21,000	-5	6
A-9	Reach-in Ice Cream	22,400	-12	12.6
A-10	Reach-in Ice Cream	19,200	-12	10.8
A-11	Reach-in Ice Cream	25,600	-12	14.4
A-12	Reach-in Ice Cream	22,400	-12	12.6
A-13	Bakery Freezer	9,400	-5	4
A-14	Deli Freezer	9,400	-5	4
	Pump Heat	11,400		
	Total Refrigeration Load	281,300		144
Brine Supply Temperature = -20°F Chiller SST = -25°F				

Table 4. Medium Temperature Circuits for the Secondary Loop Refrigeration Store (Loop B1)

Circuit	Fixture Description	Refrig Load (Btuh)	Disch. Air Temp (°F)	Brine Flow (gpm)
B1-1	Deli Walk-in Cooler	8,600	30	2.5
B1-2	Multi-deck Lunch Meat Cases	50,200	30	15.7
B1-3	Meat Holding Box	6,300	30	2
B1-4	Service Seafood Case	5,000	26	2.4
B1-5	Single-deck Meat Cases	3,300	26	1.6
B1-6	Single-deck Meat Cases	11,600	26	5.6
B1-7	Multi-deck Meat Cases	37,800	28	19.6
B1-8	Meat Walk-in Cooler	28,000	30	16
B1-9	Island Cake Display Case	5,000	28	6.9
B1-10	Self-service Bakery Case	5,000	28	5
B1-11	Multi-deck Deli Case	56,500	30	17.6
B1-12	Multi-deck Deli Case	44,000	30	13.7
B1-13	Multi-deck Deli Case	25,100	30	7.8
B1-14	Island Cheese Case	7,200	28	10.3
B1-15	Island Prepared Foods Case	7,200	28	10.3
B1-16	Service Deli Cases	6,400	31	8
B1-17	Sandwich Cases	14,600	28	13.5
	Pump Heat	16,000		
	Total Refrigeration Load	337,800		159
Brine Supply Temperature = 20°F Chiller SST = 15°F				

Table 5. Medium Temperature Circuits for the Secondary Loop Refrigeration Store (Loop B2)

Circuit	Fixture Description	Refrig Load (Btuh)	Disch. Air Temp (°F)	Brine Flow (gpm)
B2-1	Flower Walk-in Cooler	8,600	38	4
B2-2	Melon Table	6,400	35	3.8
B2-3	Multi-deck Produce Cases	37,600	36	23.8
B2-4	Multi-deck Produce Cases	34,200	36	21.6
B2-5	Multi-deck Produce Cases	9,000	36	3.2
B2-6	Multi-deck Produce Cases	9,000	36	3.2
B2-7	Multi-deck Produce Cases	36,000	36	12.7
B2-8	Multi-deck Produce Cases	27,400	36	17.3
B2-9	Produce Walk-in Cooler	23,900	38	14
B2-10	Meat Prep Room	45,800	50	12
B2-11	Multi-deck Egg Case	12,000	35	4.2
B2-12	Dairy Walk-in Cooler with 14 Doors	46,200	36	27
B2-13	Bakery Walk-in Cooler	7,300	36	4.5
B2-14	Multi-deck Beverage	56,600	36	20.1
B2-15	Multi-deck Beverage	44,700	36	15.9
B2-16	Floral Case	7,800	36	4
B2-17	Pasta Case	12,000	31	4.3
	Pump Heat	21,800		
	Total Refrigeration Load	446,300		196
	Low Temp Subcooling	49,800		
Brine Supply Temperature = 25°F Chiller SST = 20°F				

Figure 5 and Figure 6 show diagrams describing the operation of the secondary loop refrigeration system for low and medium temperature, respectively. The refrigeration load is divided into 3 secondary fluid circuits operating at -20, 20, and 25°F, respectively. There are three compressor suction groups to provide refrigeration to the secondary fluid loops operating at saturated suction temperatures of 25, 15, and 20°F. Multiple parallel compressors are employed in each of the suction groups so that compressors can be cycled on and off to provide capacity control by maintaining a constant suction pressure. The compressed gas is routed through two discharge manifolds; one manifold transports the gas from the -20°F compressors while the seconds carries the combined flow from the 15 and 20°F compressors. The compressor discharge gas may be used for heat reclaim for either water or space heating, or for defrost brine heating. The discharge refrigerant flow is piped to the evaporative condenser. From the condenser, the liquid refrigerant flow from the two discharge circuits is combined at a single receiver tank. The liquid refrigerant flow from the receiver is divided into two streams, which pass through heat exchangers that subcool the refrigerant by warming brine that is used for display case defrost. After these heat exchangers, the liquid refrigerant flow is combined at the liquid manifold where it is distributed to the three chiller evaporators. The only exception is the liquid refrigerant flow used at the low temperature (-25°F) chiller evaporator. The refrigerant liquid for the low temperature system is further sub-cooled in a mechanical sub-cooling heat exchanger. The refrigeration for the mechanical subcooling is provided by the 20°F suction group compressors.

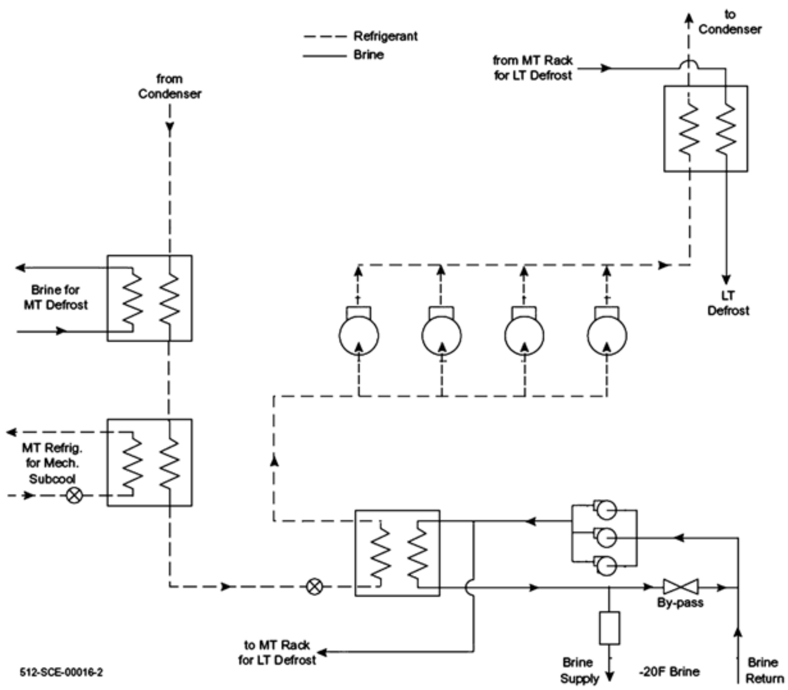

Figure 5. Piping Diagram for the Low Temperature Secondary Loop Refrigeration System (Patent by Hill PHOENIX, Inc.)

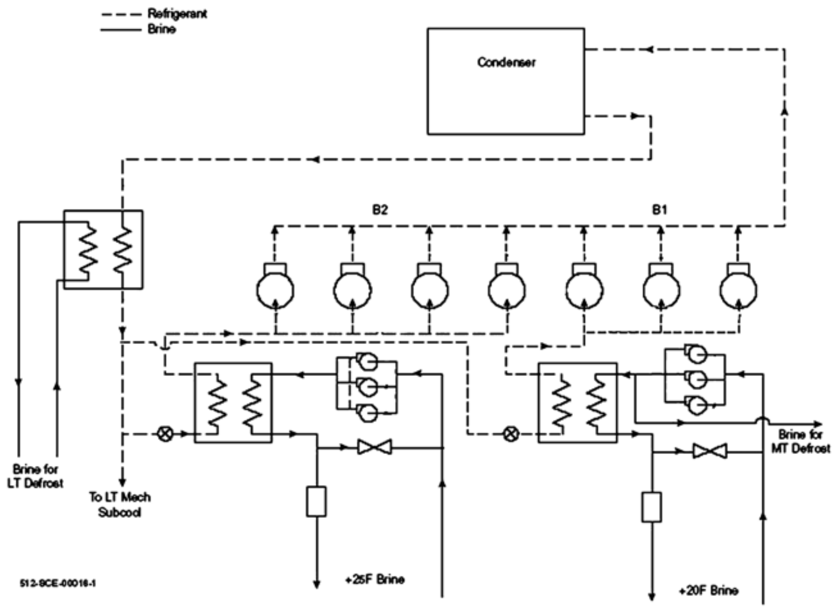

Figure 6. Piping for the Medium Temperature Secondary Loop Refrigeration System (Patent by Hill PHOENIX, Inc.)

In each fluid loop, the secondary fluid is circulated between the chiller evaporator and the refrigerated fixtures by three parallel centrifugal pumps. The number of pumps operating is determined by the pump discharge pressure. Pumps are cycled off on a rise in discharge pressure and restarted when the pressure falls.

2.3.2 Measurement Plan to Monitor Secondary Loop Refrigeration

A measurement plan was developed to determine the necessary instrumentation to evaluate the performance of the secondary loop refrigeration. The key elements included in the evaluation were the following:

- Refrigeration system energy consumption
- Refrigeration supplied
- Operating state points
- Display case operating temperatures
- Ambient temperature and humidity

Details of these measurements are as follows:

2.3.2.1 Refrigeration Energy Consumption

Since energy efficiency is of primary concern to the project, power measurement of the refrigeration system was the primary measurement monitored. Power measurements of refrigeration compressors, brine pumps, and condenser fans and pumps were noted at regular intervals and used to determine energy consumption of the refrigeration system.

Power measurements were recorded at 10-second intervals which were later combined into 15-minute averages. The average values were stored at the data acquisition for remote collection. Watt transducers were employed for these measurements.

Total store power and electric energy were also monitored and recorded.

2.3.2.2 Refrigeration Supplied

Differences in the refrigeration loads addressed at the two sites (Secondary and Multiplex), must be taken into account when comparing energy use of the two systems. Refrigeration supplied by the secondary loops and the chiller systems were calculated based upon several instrument readings taken at regular intervals. The refrigeration supplied by each brine loop was determined from:

$$\mathrm{Q_{brine}} = m\,c\left(\mathrm{T_{bin}} - \mathrm{T_{bout}}\right)$$

where

> T_b is the temperature of the brine in supply and return lines at the chiller evaporator
> m is the mass flow rate of brine in the loop
> c is the specific heat of the brine

The refrigeration load delivered by the refrigeration to the chiller evaporator is found from:

$$\mathrm{Q_{chiller}} = m\left(h_{out} - h_{liq}\right)$$

where

> h is the enthalpy of the refrigerant at the outlet (out) and inlet (liq) of the chiller
> m is the mass flow rate of refrigerant entering the chiller.

A third method was available to estimate the refrigeration provided by using the published capacity data for the compressors. The rated capacity of each compressor is given as an equation of the saturated suction and discharge temperatures at which the compressor is operating. Corrections are needed to account for the refrigerant temperature at the suction of the compressor and for the refrigerant liquid. The rated capacity was calculated with measured data for these quantities. The refrigeration load was found by multiplying the capacity by the operating run fraction for each compressor. Monitoring and recording on/off cycles of each compressor determined the run fraction.

The results from the power and refrigeration measurements were then combined to determine the energy efficiency ratio (EER) of the refrigeration system. The EER is defined as:

$$EER = \frac{Refrigeration Supplied}{Power Required} (Btu/Watt - hr)$$

For the secondary loop refrigeration, the power required included both the compressor and brine pumping power. Two EER values were calculated, one for low temperature refrigeration and a second for medium temperature refrigeration. The medium temperature EER combined the refrigeration and power of the 15 and 20°F fluid loops. Refrigeration provided by the 20°F loop for mechanical subcooling was subtracted from the medium temperature refrigeration load.

2.3.2.3 Refrigeration System Operating State Points

The significant operating state points that were monitored for the secondary loop system are the following:

- Saturated Suction Temperature (SST) – This is the saturation temperature corresponding to the refrigerant pressure measured at the compressor suction.
- Saturated Discharge Temperature (SDT)– The saturation temperature of the refrigerant based upon the pressure measured at the compressor discharge.
- Return Gas Temperature – The temperature of the refrigerant gas measured at the suction of the compressor.
- Refrigerant Liquid temperature – The liquid temperature was measured at several locations in the system to characterize the subcooling provided. The liquid temperature was measured at the outlets of the receiver and ambient subcooling coil, and at the inlet and outlet of the defrost heat exchanger. Additional liquid temperature measurements were needed at the outlet of the mechanical subcooling heat exchanger for the low temperature chiller.
- Refrigerant discharge temperature – The temperature was measured at the inlet of the brine defrost heat exchanger, and at the inlets of the heat reclaim water heater (an outlet temperature was also required) and space heater coil.
- Condenser Pan Water temperature – the primary control point for the condenser fans.
- On/off digital signals were included to track events such as the initiation of defrost or heat reclaim.

2.3.2.4 Brine Pump Operation

The operation of the brine pumps was monitored to evaluate the pump control strategy. Along with pump power, digital on/off signals were included to monitor the operation of the brine pumps.

2.3.2.5 Brine Defrost Operation

The warm brine defrost system specified in this project used a combination of refrigerant liquid subcooling and discharge heat to raise the brine temperature. The subcooling of the refrigerant increases the refrigeration capacity of the compressors and is an energy-saving feature for this system. Operation of the hot brine defrost was fully characterized as part of the site monitoring. The following measurements were included for each secondary loop:

- Brine flow rate for defrost – a flow meter was installed in each brine defrost supply line.
- Brine temperature in and out of defrost heat exchanger.
- Brine temperature in and out of defrost/subcooling heat exchanger.

2.3.2.6 Display Case Operation

The operation of the display cases and walk-in boxes was monitored during testing. "Degree Master" modules were employed to control operation of the display cases. These modules regularly measured discharge and return air temperatures, along with brine inlet and outlet temperatures.

Two display case circuits were also equipped with product simulators in each display case that allow measurement and tracking of product temperature. The 2 circuits selected consisted of:

- Single-deck meat cases –three display cases
- Multi-deck produce cases –two display cases

Similar display cases at the multiplex store were also equipped with product simulators to allow comparison of product temperature and temperature variation for the two sites.

Other case measurements that were addressed included the energy use for case lights, fans, and anti-sweat heaters (frozen food door cases). Power monitoring was included for this purpose.

2.3.2.7 Ambient Conditions

Both inside and outside ambient dry-bulb temperature and humidity were monitored. The inside reading consisted of multiple readings of the dry-bulb temperature and a single reading of the inside relative humidity. The outside ambient instrumentation consisted of single readings of the dry-bulb temperature and the relative humidity.

Problems were incurred with the outside humidity reading shortly after the start of testing. An alternate approach for outside ambient data was employed after the sensor failed. Southern California Edison maintains weather stations throughout the service territory. One station is located in Moorpark, CA, which is in close proximity to the secondary loop store. Good agreement was found between dry-bulb temperature and relative humidity readings from the weather station and those taken at the store during a site visit. Weather station readings were used for the entire test period to represent the outside ambient conditions at the store. Similarly, weather station readings were also used to characterize the ambient for the multiplex test store. Both the store and the weather station are located in Valencia, CA.

2.3.2.8 Data Collection at the Secondary Loop Store

All instrument readings at the secondary loop store were wired to the energy management system (EMS) at the site. The instruments were read by the EMS at two-minute intervals and these readings were stored for later retrieval. Data were obtained from the EMS by a modem/phone line connection. The site was called daily and the data were downloaded.

2.3.3 Field Testing of the Multiplex Refrigeration System

2.3.3.1 Description of the Multiplex Refrigeration Test Site

A second Vons supermarket, located in Valencia, CA, was selected and instrumented for performance comparison with the secondary loop refrigeration system. The store selected employed a direct expansion, multiplexed refrigeration system to provide refrigeration to all display cases and walk-in coolers. The total building area of the store is 56,526 ft^2. Figure 7 shows the layout of the sales area of the store, which measured 41,912 ft^2. Table 6, Table 7, Table 8, and Table 9 describe the refrigerated fixtures at the multiplex test store.

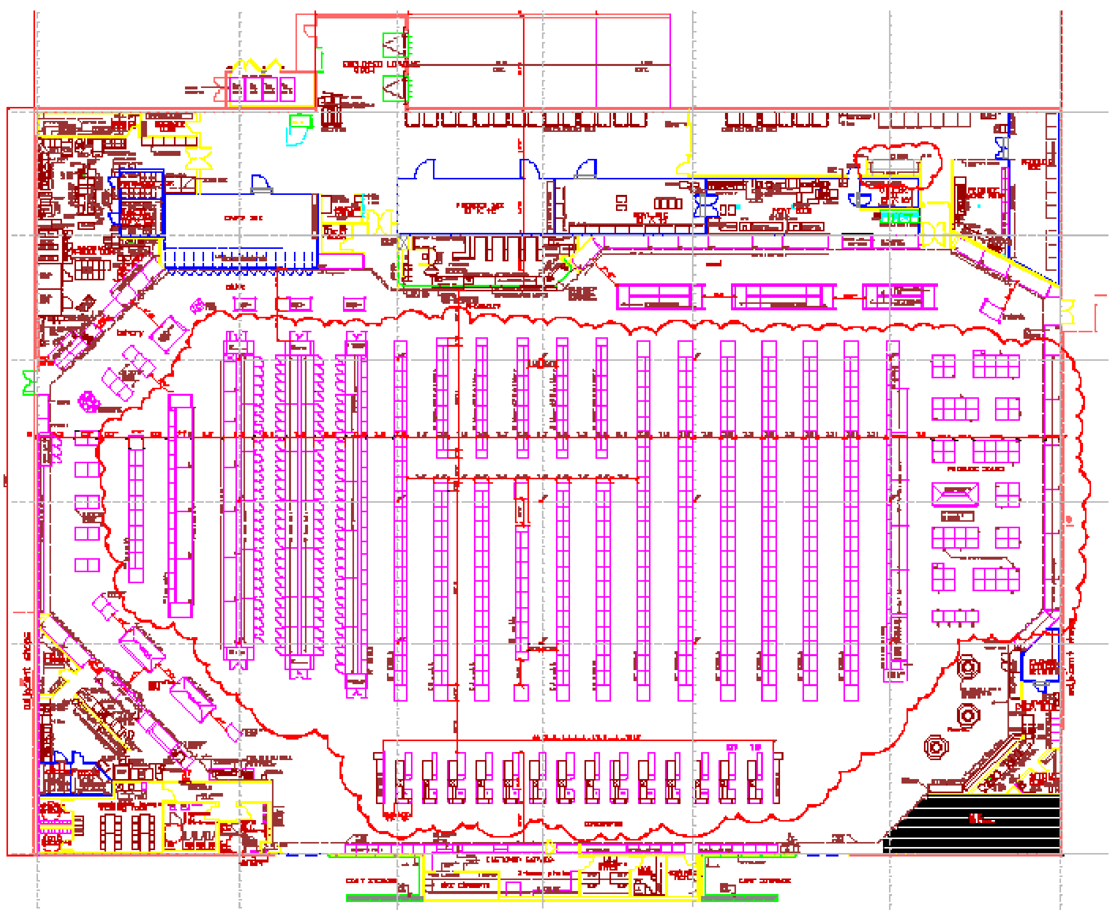

Figure 7. Layout of the Multiplex Refrigeration Test Store

Table 6. Low Temperature Circuits at the Multiplex Refrigeration Store (Rack A)

Circuit	Fixture Description	Refrig Load (Btuh)	Evap Temp (°F)	Disch Air Temp (°F)
A1	Reach-in Ice Cream/Frozen Food Case	28,050	-19	-12
A2	Reach-in Ice Cream/Frozen Food Case	28,050	-19	-12
A3	Reach-in Ice Cream/Frozen Food Case	29,700	-19	-12
A4	Reach-in Ice Cream/Frozen Food Case	28,050	-19	-12
A5	Reach-in Ice Cream/Frozen Food Case	26,400	-19	-12
A6	Reach-in Ice Cream/Frozen Food Case	24,750	-19	-12
A7	Reach-in Ice Cream/Frozen Food Case	28,050	-19	-12
A8	Reach-in Ice Cream/Frozen Food Case	33,000	-19	-12
A9	Wide Island Frozen Fish/Meat Case	17,050	-20	-12
A10	Wide Island Frozen Fish/Meat Case	13,950	-20	-12
A11	Deli Freezer	8,200	-15	-5
A12	Bakery Freezer	10,000	-15	-5
A13	Grocery Freezer	35,000	-20	-10
A14	Reach-in Ice Cream/Frozen Food Case	4,950	-19	-12
	Total Refrigeration Load (Btuh)	315,200		
Low Temp Rack A SST = -25°F				

Table 7. Medium Temperature Circuits at the Multiplex Refrigeration Store (Rack B)

Circuit	Fixture Description	Refrig Load (Btuh)	Evap Temp (°F)	Disch Air Temp (°F)
B2	Deli Walk-in Cooler	5,800	20	36
B4	Sandwich Case	5,600	18	28
B1	Deli Island Case	14,400	15	25
B3	Cheese Island Case	14,400	15	25
B5	Service Deli Cases	7,040	15	28
B6	Multi-deck Deli Cases	22,800	24	32
B7	Bakery Island Cases	9,600	15	25
B8	Retarder Box	8,000	20	36
B9	Self-Service Bakery Case	26,800	20	34
B10	Multi-deck Dairy Case	17,100	24	32
B18	Produce Island Case	14,400	15	25
B11	Multi-deck Meat Cases	45,720	21	29
B12	Single-deck Meat Cases	12,960	21	28
B13	Self-Service Fish Case	5,448	19	28
B14	Service Fish Cases	2,400	15	
B15	Wide Island Meat Cases	22,680	18	26
B16	Multi-deck Lunch Meat Cases	39,900	24	32
B17	Multi-deck Lunch Meat Cases	11,400	24	32
	Total Refrigeration Load (Btuh)	286,448		
Medium Temp Rack B SST = 13°F				

Table 8. Medium Temperature Circuits at the Multiplex Refrigeration Store (Rack C)

Circuit	Fixture Description	Refrig Load (Btuh)	Evap Temp (°F)	Disch Air Temp (°F)
C1	Multi-deck Dairy/Beverage Cases	74,100	24	32
C2	Multi-deck Beverage Cases	108,300	24	32
C3	Multi-deck Produce Cases	77,600	21	31
C4	Multi-deck Produce Cases	42,750	24	32
C5	Single-deck Produce Cases	46,560	21	31
C6	Floral Walk-in Cooler	8,100	20	38
C11	Floral Display Case	12,200	20	38
C7	Produce Walk-in Cooler	39,120	20	36
C8	Seafood Walk-in Cooler	9,700	20	36
C9	Dairy Walk-in Cooler with 16 Doors	50,000	20	32
C10	Meat Walk-in Cooler	31,400	20	32
	Total Refrigeration Load (Btuh)	499,830		
Medium Temp Rack C SST = 20°F				

Table 9. Medium Temperature Circuits at the Multiplex Refrigeration Store (Rack D)

Circuit	Fixture Description	Refrig Load (Btuh)	Evap Temp (°F)	Disch Air Temp (°F)
D1	Produce Prep Area	24,000	37	50
D2	Meat Prep Area	95,000	37	50
	Total Refrigeration Load (Btuh)	119,000		
	Subcooling System A	49,950		
	Subcooling System B	44,708		
	Subcooling System C	81,634		
	Total Subcooling Load (Btuh)	176,292		
Medium Temp Rack D SST = 35°F				

Figure 8 and Figure 9 show diagrams of the multiplex refrigeration system at the test store, for the low and medium temperature refrigeration, respectively. The refrigeration was divided over three compressor racks, consisting of one low temperature and two medium temperature racks. One of the medium temperature racks (labeled as C/D in the diagram) has two suction pressure groups. The highest temperature suction group, D, was also used for mechanical subcooling of the remaining low and medium temperature racks. The arrangement is considered unusual in that normally only the low temperature rack is mechanically subcooled. An evaporative condenser provided heat rejection. The refrigeration system was equipped to provide heat reclaim for both hot water and space heating.

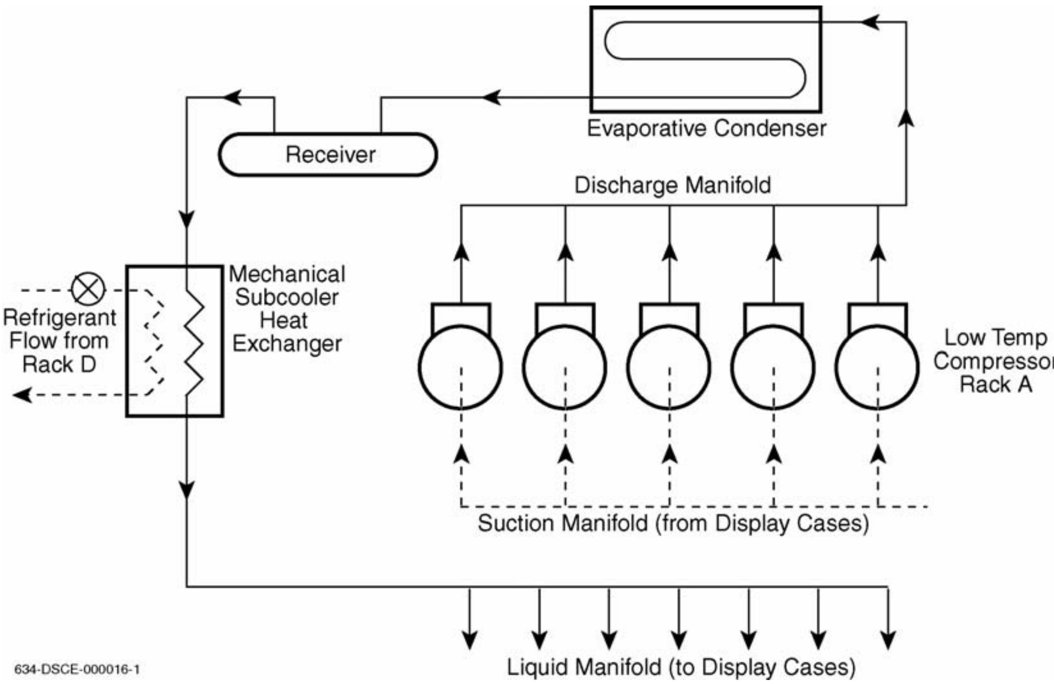

Figure 8. Piping Diagram for the Low Temperature Multiplex Refrigeration

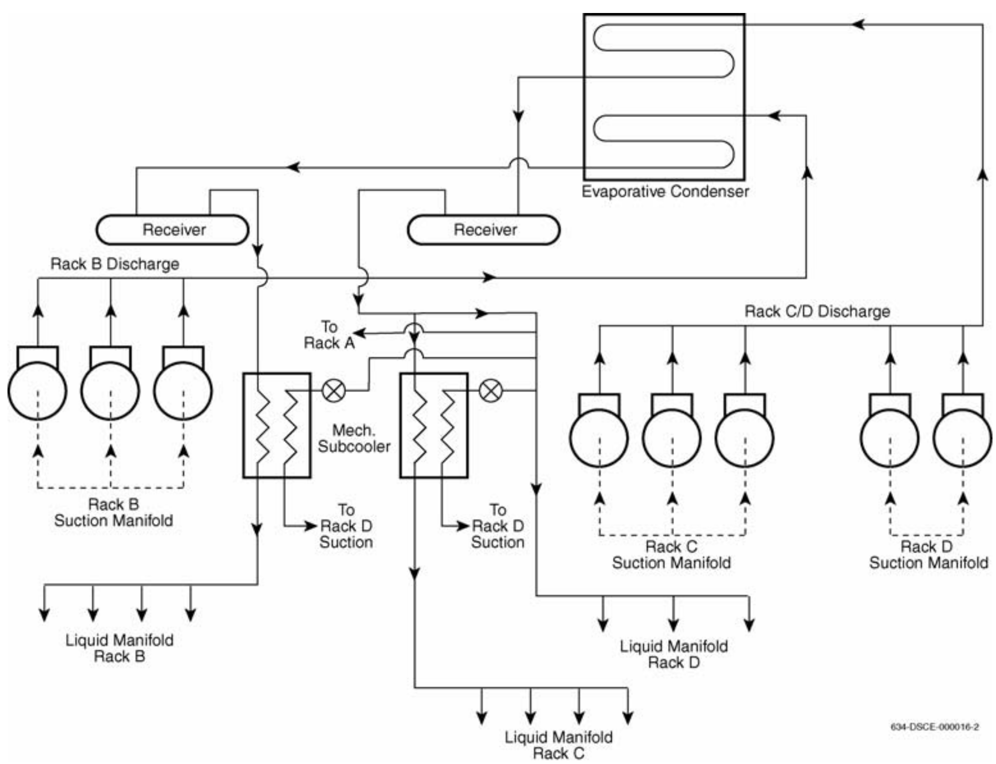

Figure 9. Piping Diagram for the Medium Temperature Multiplex Refrigeration

2.3.4 Measurement Plan for the Multiplex Refrigeration System

A measurement plan was also developed for the multiplex system that included all key parameters to allow comparison of performance with that of the secondary loop refrigeration. The elements needed for this comparison include the following:

- Refrigeration system energy consumption
- Refrigeration supplied
- Operating state points
- Display case operating temperatures
- Ambient temperature and humidity

2.3.4.1 Refrigeration Energy Consumption

Power measurements of refrigeration compressors, and condenser fans and pumps were recorded from watt transducers at regular intervals. The measurements were used to determine energy consumption and for the multiplex refrigeration system EER calculation.

The total power and energy of the store was also monitored.

Refrigeration Supplied

The refrigeration load delivered by each refrigeration rack was found from:

$$Q_{rack} = \dot{m}\left(h_{out} - h_{liq}\right)$$

where

h is the enthalpy of the refrigerant at the suction of the compressor rack (out) and at the liquid manifold (liq).
m is the mass flow rate of refrigerant circulated by the compressors.

The refrigeration provided was also estimated through the published capacity data for the compressors and digital on/off readings that described compressor operation as explained above for the secondary loop compressors. Several of the medium temperature compressors employed cylinder unloading for capacity control. Operation of the unloaders was monitored and counted by digital on/off readings.

2.3.4.2 Refrigeration System Operating State Points

The significant operating state points that were monitored for each multiplex compressor rack and for each compressor suction group are the following:

- Saturated Suction Temperature (SST) – the saturation temperature corresponding to the refrigerant pressure measured at the compressor suction.

- Saturated Discharge Temperature (SDT) – The saturation temperature of the refrigerant based upon the pressure measured at the compressor discharge.
- Return gas temperature – The temperature of the refrigerant gas measured at the suction of the compressor.
- Refrigerant liquid temperature – The liquid temperature was measured at the outlet of the receiver and before and after each mechanical subcooling heat exchangers to characterize the subcooling provided.
- Refrigerant discharge temperature – Measured at the inlets of the heat reclaim water heater (an outlet temperature was also required) and space heater coils.
- Condenser pan water temperature – the primary control point for the condenser fans.
- On/off digital signals were used to track events such as the initiation of defrost or heat reclaim.

2.3.4.3 Display Case Operation

The operation of the display cases and walk-in boxes were monitored during testing. Each display case is equipped with a discharge air temperature reading used to control operation of the refrigeration system.

Several display cases were equipped with product simulators, similar to those used with the secondary loop system, so an evaluation of product temperature maintenance could be made. The display cases equipped with product simulators were

- Single-deck meat – 2 display cases
- Multi-deck produce – 2 display cases

The display cases selected for product simulators were similar in type and operated at the same discharge air temperature as those cases instrumented at the secondary loop store.

Other case measurements that were addressed included power readings for case lights, fans, and anti-sweat heaters (frozen food case doors).

2.3.4.4 Ambient Conditions

Inside ambient conditions were monitored with several dry-bulb temperature measurements and one relative humidity reading. The outside ambient conditions were determined through dry-bulb temperature and relative humidity readings taken at a Southern California Edison weather station also located in Valencia, CA.

2.3.5 Data Collection at the Multiplex Refrigeration Test Store

All instrumentation used for the evaluation of the multiplex refrigeration system was wired to the store EMS, where readings were taken at two-minute intervals and stored. The EMS was polled on a daily basis through a modem and phone line to gather the data. The weather station data were collected routinely by Southern California Edison and provided to the project on a monthly basis.

3.0 Project Outcomes

3.1 Analysis Results

The analytical models were used to predict the annual energy consumptions of the secondary loop and multiplex refrigeration systems. The design refrigeration loads for the secondary loop store were used to estimate the energy consumptions of both systems. Weather data for Moorpark, CA for 2001 were used to generate the ambient temperature bins for the analysis.

Table 10 provides the annual energy estimates for the secondary loop refrigeration system. The modeled secondary loop system included evaporative condensing, multiple parallel secondary fluid pumping, and warm brine defrost subcooling to help decrease energy consumption. Mechanical subcooling was modeled for the low temperature refrigeration. Table 11 has the energy estimates for the multiplex system with air-cooled condensing, while Table 12 shows the energy consumption for the multiplex system with evaporative condensing. The multiplex refrigeration system employed mechanical subcooling for the low temperature refrigeration only.

Table 10. Secondary Loop Modeling Results

System	Brine Loop Temp	Design Load	Annual Energy (kWh)			
	°F	Btuh	Compressors	Pumping	Condenser	Total
Low Temp	-20	281,300	263,951	19,270	31,863	315,084
Med Temp 1	20	337,800	169,977	22,406	39,938	232,321
Med Temp 2	25	446,300	214,914	25,598	45,628	286,140
Subcooling		49,750				
Total			648,842	67,328	117,429	833,599

Table 11. Multiplex Modeling Results – Air-Cooled Condensing

System	SST	Design Load	Annual Energy (kWh)		
	°F	Btuh	Compressors	Condenser	Total
Low Temp	-23	281,300	304,235	70,418	374,653
Med Temp 1	13	337,800	216,580	25,658	242,238
Med Temp 2	19	446,300	270,676	91,906	362,582
Subcool		49,750			
Total			791,491	187,982	979,473

Table 12. Multiplex Modeling Results – Evaporative Condensing

System	SST	Design Load	Annual Energy (kWh)		
	°F	Btuh	Compressors	Condenser	Total
Low Temp	-23	281,300	290,823	29,965	320,788
Med Temp 1	13	337,800	186,366	21,836	208,202
Med Temp 2	19	446,300	231,637	79,102	310,739
Subcool		49,750			
Total			708,826	130,903	839,729

The results of the analysis show that the secondary loop refrigeration system had lower energy consumption than either of the multiplex system configurations. Annual energy savings achieved by the secondary loop system versus the multiplex system with air-cooled condensing were 145,874 kWh, which were savings of 14.9%. Versus multiplex with evaporative condensing, the annual savings were 6,130 kWh, or 0.3%

3.2 Field Test Results

3.2.1 Energy Comparison

Summaries of the electric energy data collected at the two sites are given in Table 13 and Table 14 for the multiplex and secondary loop stores, respectively. Values are listed for each month of testing and are expressed as average daily energy consumptions. A breakdown is provided for the refrigeration system in terms of the rack (compressors only for the multiplex, and compressors and pumps for the secondary loop), the condenser, and the display cases. The display case energy includes the energy consumptions for fans, lights, and anti-sweat heaters. No values for store or display case energy were recorded for the secondary loop store during January, because of problems incurred with the data acquisition. Figure 10 shows the average daily energy values for the store and for total refrigeration for both the multiplex and secondary loop sites. The secondary loop refrigeration system represents a larger percentage of the total store energy, because the secondary loop site is smaller than the multiplex site and has lower total energy use.

Table 13. Multiplex Refrigeration Average Daily Energy Consumption

	Average Daily Energy (kWh/day)					
Month	Compressors	Condenser	Cases	Refrigeration	Store	% of Store
Oct '02	1,994	189	726	2,909	6,530	44.5
Nov	1,871	157	708	2,736	6,609	41.4
Dec	1,920	132	961	3,013	7,021	42.9
Jan '03	2,039	157	799	2,995	7,066	42.4
Feb	1,989	126	764	2,879	6,851	42.0
Mar	2,030	144	732	2,906	6,396	45.4
Apr	2,072	131	739	2,942	6,471	45.5
May	2,016	302	760	3,078	6,446	47.8
June	2,183	297	781	3,261	6,822	47.8
Overall	2,013	182	774	2,969	6,690	44.4

Table 14. Secondary Loop Refrigeration Average Daily Energy Consumption

	Average Daily Energy (kWh/day)						
Month	Compressors	Pumps	Condenser	Cases	Refrigeration	Store	% of Store
Oct '02	2,069	191	299	1,140	3,699	6,265	59.0
Nov	1,952	187	253	1,103	3,495	6,134	57.0
Dec	1,901	188	199	1,113	3,401	6,034	56.4
Jan '03	1,937	201	249	NR	2,387	NR	-
Feb	1,853	213	208	1,116	3,390	5,968	56.8
Mar	2,035	204	243	1,083	3,565	6,131	58.1
Apr	2,038	195	243	1,127	3,603	6,147	58.6
May	2,151	195	309	1,061	3,716	6,281	59.2
June	2,303	203	316	988	3,810	6,369	59.8
Overall	2,027	197	258	1,091	3,452	6,166	58.0

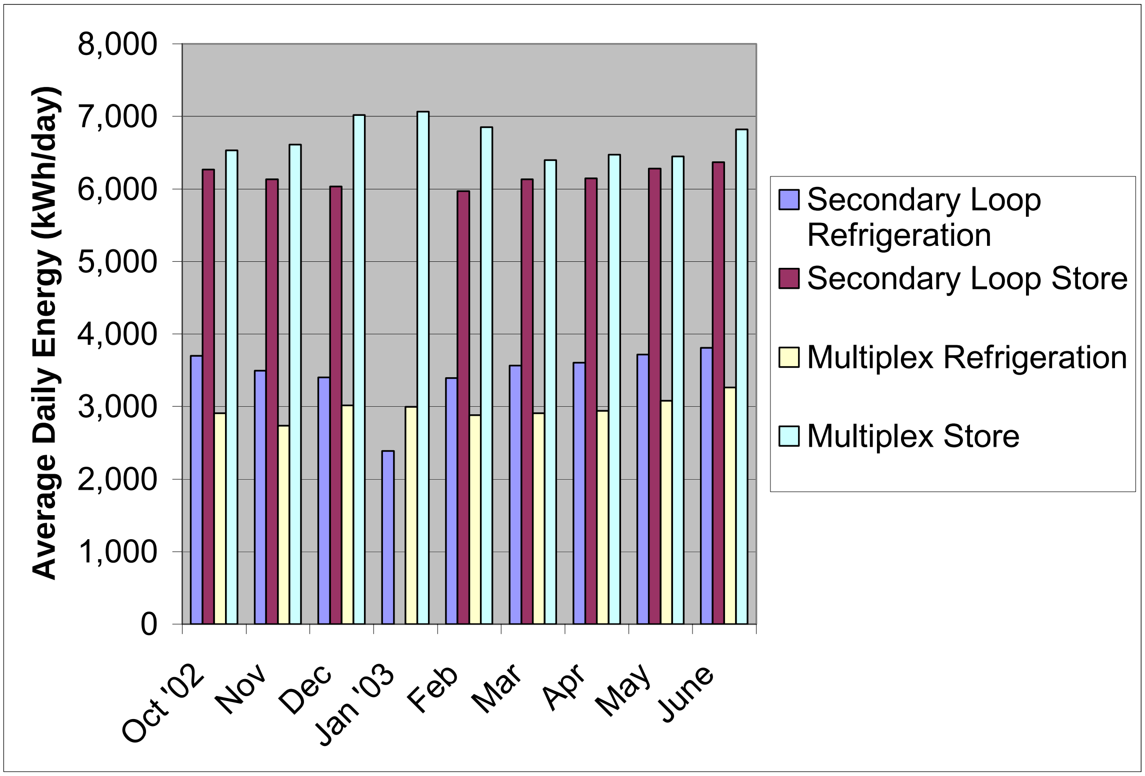

Figure 10. Store and Total Refrigeration Energy use for the Multiplex and Secondary Loop Test Sites

Figure 11 and Figure 12 show the breakdowns of energy use for refrigeration for the multiplex and secondary loop systems. Compressor energy was the largest component at each site followed by display case energy. The compressor energy at the secondary loop site was approximately the same as the multiplex system. Both display case and condenser energy was higher at the secondary loop site. One reason for the higher case energy use was the display case lights were turned off at the multiplex site during late evening hours; this practice was not followed at the secondary loop store. The condenser energy use at the multiplex store was lower than at the secondary loop store. The multiplex condenser was sized to handle heat rejection for both the refrigeration and store air conditioning, which meant the condenser was over-sized for refrigeration operation alone. The air conditioning ran only briefly during the test period, meaning minimum fan operation was required to maintain the condenser minimum set point temperature.

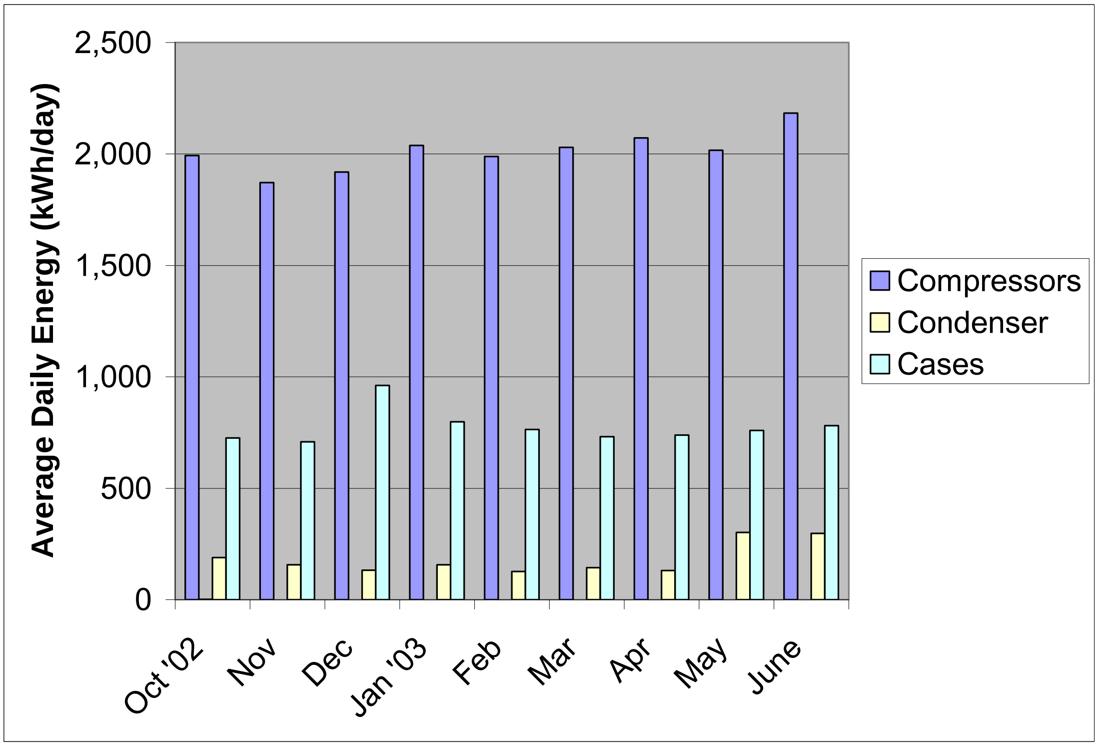

Figure 11. Multiplex Refrigeration System Energy Consumption

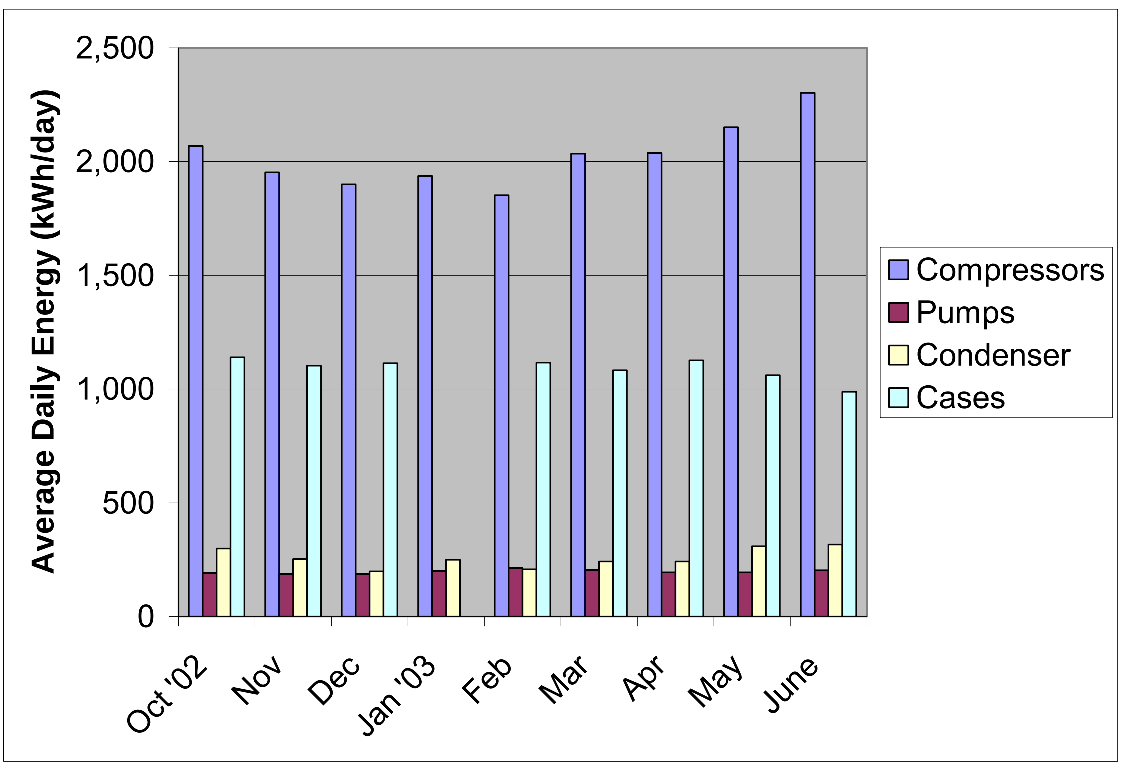

Figure 12. Secondary Loop Refrigeration Energy Consumption

One of the major energy differences between the multiplex and secondary loop refrigeration systems was the operation of the secondary loop pumps. To minimize the pump energy, multiple parallel pumps were installed on each fluid loop; these pumps were multiplexed based upon the pressure difference across the pumps. Table 15 shows the overall results achieved by this arrangement. The pump run fractions are shown for each of the fluid loops, along with the average daily pump energy consumed. Maximum daily pump energy was then calculated, which represents the energy consumed if all pumps ran continuously. This maximum consumption would occur if the fluid loops were operated at a constant flow rate at all times. The energy savings are the difference between the maximum and average consumptions. The use of the multiple pump arrangement resulted in an energy savings of 58%. The savings of 273.2 kWh/day are 7.9% of the energy consumption of the secondary loop refrigeration system.

Table 15. Performance of Multiple Pump Arrangement for Secondary Fluid Pumping

	Pump Run Fraction			Pump Energy (kWh/day)		
Fluid Loop	Pump 1	Pump 2	Pump 3	Avg. Pump Energy	Max. Pump Energy	Savings
Low Temp	0.99	0.42	0.00	83.3	177.2	93.9
Med. Temp 1	0.99	0.33	0.05	60.0	131.4	71.4
Med. Temp 2	0.99	0.01	0.00	54.0	161.9	107.9
Overall				197.3	470.5	273.2

3.2.2 Power Demand

Table 16 and Table 17 give the peak power demand (15 min.) recorded for each test month for the multiplex and secondary loop stores, respectively. The day and time of the peak are also noted in the tables. The power for the total refrigeration that occurred at the same time is also shown, along with the contribution of the refrigeration to the peak demand. No store demand was measured for the secondary loop store during January because of data acquisition problems. Figure 13 shows a graphical representation of these power values. The peak power demand of the multiplex store was higher than the secondary loop site

Table 16. Peak Power Demand of the Multiplex Store

	Date	Time	Store	Refrig	% of Store
Oct '02	9-Oct	16:55	370.3	135.2	46.2
Nov	9-Nov	15:39	329.3	118.6	47.3
Dec	16-Dec	15:47	348.6	102.3	41.5
Jan '03	16-Jan	8:31	335.2	102.2	42.9
Feb	11-Feb	11:13	351.9	117.7	43.4
Mar	14-Mar	14:29	324.3	126.2	50.2
Apr	4-Apr	9:50	322.7	100.1	41.5
May	28-May	14:37	332.7	133.5	50.9
Jun	16-Jun	14:22	341.9	129.3	48.5

Table 17. Peak Power Demand for the Secondary Loop Store

	Date	Time	Store	Refrig	% of Store
Oct '02	6-Oct	16:30	312	111.2	51.5
Nov	9-Nov	17:15	312	124.8	56.7
Dec	4-Dec	17:15	304	124.8	56.6
Jan '03	NR				
Feb	13-Feb	9:45	296	120	57.6
Mar	31-Mar	12:00	320	130.4	56.5
Apr	16-Apr	13:30	320	146.4	62
May	19-May	12:00	320	136.8	57.3
Jun	16-Jun	16:00	320	150.4	60.5

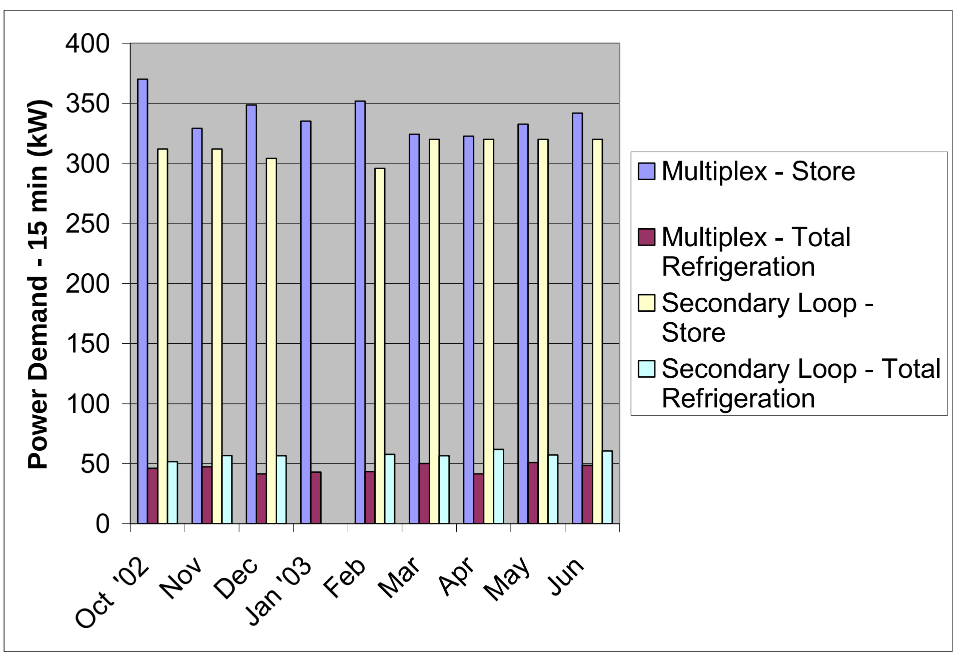

Figure 13. Peak Power Demand for the Multiplex and Secondary Loop Stores

3.2.3 Refrigeration Efficiency

Table 18 and Table 19 show the average refrigeration loads, power, and EER for the multiplex and secondary loop systems, respectively. The refrigeration load at each site has been divided into two segments which are the low and medium temperature refrigeration loads. The low temperature refrigeration load is addressed by the low temperature system at each site. The remainder of the refrigeration load was combined as the medium temperature load. The compressor power was also divided on this basis. The only exception is the mechanical subcooling load, which is subtracted from the medium temperature refrigeration.

For the multiplex refrigeration, the power and EER were also determined if no mechanical subcooling were applied to the medium temperature refrigeration. Mechanical subcooling is not normally used with medium temperature refrigeration and would not be representative of a baseline multiplex system found in most supermarkets. An estimate of the power level expected if the medium temperature racks were not subcooled is also given to provide an additional comparison with the secondary loop system. The estimate performance for non-subcooled compressors was made by first determining the total refrigeration capacity of the medium temperature compressors using the measured saturated suction and discharge temperatures along with the measured return gas and liquid temperatures, and the performance curves for the compressors. The total run fraction for the compressors was estimated from this capacity and the refrigeration load measured at the site:

$$NoSubRunFraction = \frac{Q_{ref}}{\sum Cap}$$

where

Q_{ref} is the measured refrigeration load
Σ Cap is the total non-subcooled capacity of the compressors

The estimated power of the compressors is then found by multiplying the combined power for the compressors with the *NoSubRunFraction*.

Table 18. Multiplex Refrigeration

	Low Temperature			Medium Temperature			Med Temp No Subcooling	
	Load (kBtuh)	Rack Power (kW)	EER (Btu/W-hr)	Load (kBtuh)	Rack Power (kW)	EER (Btu/W-hr)	Rack Power (kW)	EER (Btu/W-hr)
Oct '02	323.2	42.2	7.65	554.6	44.0	12.60	46.2	12.00
Nov	316.7	40.9	7.73	508.3	39.5	12.87	41.4	12.28
Dec	344.6	43.4	7.94	479.1	36.7	13.06	38.3	12.51
Jan '03	333.2	43.1	7.73	540.3	41.9	12.90	44.0	12.28
Feb	321.9	41.6	7.74	526.1	41.0	12.82	43.2	12.17
Mar	328.3	42.6	7.71	533.9	42.4	12.58	44.6	11.96
Apr	336.5	43.0	7.81	536.9	43.3	12.39	45.5	11.79
May	336.3	43.2	7.80	514.4	44.3	11.61	46.8	11.00
Jun	337.5	43.4	7.78	582.3	47.3	12.30	49.3	11.81
Overall	331.5	42.7	7.76	530.1	42.2	12.60	44.3	12.00

Table 19. Secondary Loop Refrigeration

	Low Temperature			Medium Temperature		
	Load (kBtuh)	Rack Power (kW)	EER (Btu/W-hr)	Load (kBtuh)	Rack Power (kW)	EER (Btu/W-hr)
Oct '02	350.9	42.9	8.18	587.6	47.2	12.47
Nov	348.8	41.9	8.32	560.6	45.3	12.39
Dec	350.2	41.7	8.40	546.0	45.5	12.00
Jan '03	349.1	41.7	8.37	585.8	47.1	12.43
Feb	340.0	40.6	8.37	568.7	45.4	12.51
Mar	346.6	41.7	8.31	648.9	51.6	12.57
Apr	338.2	41.1	8.23	653.8	51.9	12.62
May	338.5	41.5	8.16	699.8	56.2	12.51
Jun	343.3	42.7	8.04	751.8	61.8	12.18
Overall	344.8	41.7	8.27	617.0	49.8	12.42

The EER values for the low temperature refrigeration show the secondary loop system operating at an approximately 6.6% higher efficiency than the multiplex system. For medium temperature refrigeration, the multiplex system EER was higher than that of the secondary loop system by 1.4%; but, when the multiplex system does not use mechanical subcooling for medium temperature, the EER of the secondary loop system is higher by 5%.

Further normalization of the refrigeration efficiency data can be achieved by considering the ambient wet-bulb temperature variation, since both refrigeration systems reject heat to an evaporative heat exchanger. Figure 14 shows the plot for the EER versus the ambient wet-bulb temperature for low temperature refrigeration. Figure 15 and Figure

16 show this same relation for medium temperature refrigeration for the multiplex and secondary loop systems, respectively. A linear regression was performed for each plot.

The normalized EER data can be used to estimate the energy consumption for each refrigeration system when addressing the same refrigeration loads and when operating at the same ambient wet-bulb temperature. For the multiplex store, the overall average refrigeration loads were 331,500 and 530,100 Btu/hr for low and medium temperature refrigeration, respectively, and the average ambient wet-bulb temperature at the multiplex site was 49.2°F. For the secondary loop store the refrigeration loads were 344,800 and 617,000 Btu/hr for the low and medium temperature, respectively, and the average ambient wet-bulb temperature was 50.3°F. Using these values, and the energy consumption of the secondary loop and multiplex refrigeration was estimated for each of the test stores. The results of this comparison are given in Table 20 and Table 21 for the multiplex and secondary loop stores, respectively. The secondary loop refrigeration used less total energy than the multiplex with either subcooled or no subcooled medium temperature refrigeration for both site descriptions.

Table 20. Comparison of Secondary Loop and Multiplex Refrigeration Using Normalized EER Values, Multiplex Store Refrigeration Loads and Ambient Wet-bulb Temperatures

	EER (Btu/W-hr)	Energy (kWh/day)	Savings (kWh/day)
Low Temperature Refrigeration			
Secondary Loop	8.29	959.7	
Multiplex	7.76	1,025.2	65.5 (6.4%)
Medium Temperature Refrigeration			
Secondary Loop	12.44	1,022.7	
Multiplex	12.60	1,009.7	-13.0 (-1.3%)
Multiplex – No Subcooling	12.01	1,059.3	36.6 (3.5%)
Total Refrigeration			
Secondary Loop		1982.4	
Multiplex		2034.9	52.5 (2.6%)
Multiplex – No Subcooling		2084.4	102.1 (4.9%)
Low Temperature Load – 331,500 Btu/hr Medium Temperature Load – 530,100 Btu/hr Average Ambient Wet-Bulb Temperature – 49.2°F			

Table 21. Comparison of Secondary Loop and Multiplex Refrigeration Using Normalized EER Values, Secondary Loop Store Refrigeration Loads and Ambient Wet-bulb Temperatures

	EER (Btu/W-hr)	Energy (kWh/day)	Savings (kWh/day)
Low Temperature Refrigeration			
Secondary Loop	8.26	1,001.2	
Multiplex	7.58	1,066.8	65.6 (6.1%)
Medium Temperature Refrigeration			
Secondary Loop	12.41	1,192.9	
Multiplex	12.54	1,181.1	-11.8 (-1.0%)
Multiplex – No Subcooling	11.94	1,240.1	47.2 (3.8%)
Total Refrigeration			
Secondary Loop		2,194.1	
Multiplex		2,247.9	53.8 (2.4%)
Multiplex – No Subcooling		2,306.9	112.8 (4.9%)
Low Temperature Load – 340,800 Btu/hr Medium Temperature Load – 617,000 Btu/hr Average Ambient Wet-Bulb Temperature – 50.3°F			

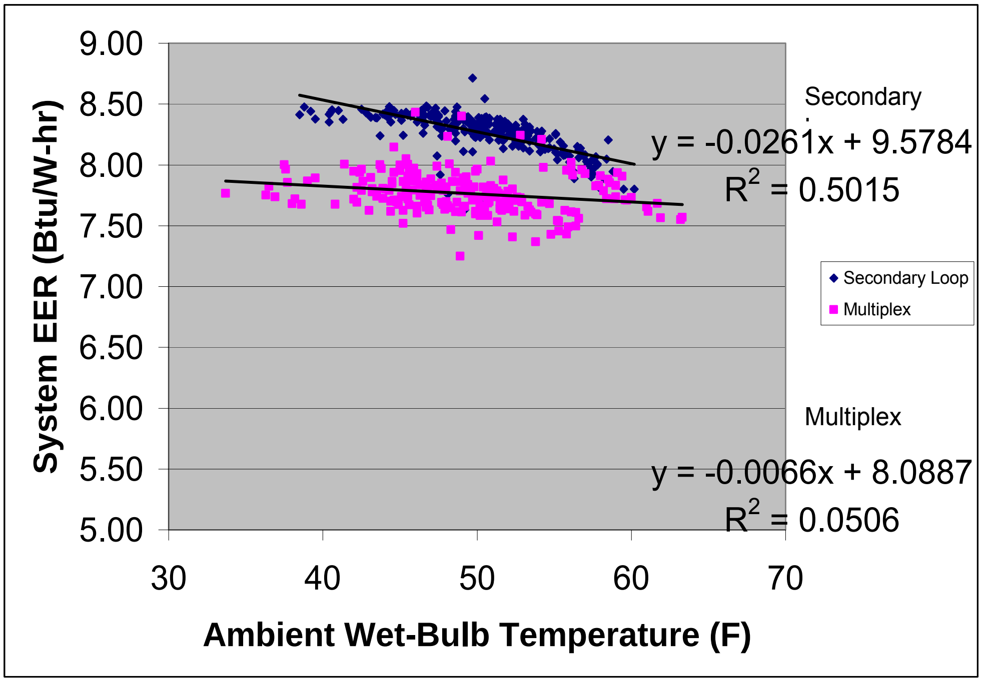

Figure 14. Relation between Ambient Wet-Bulb Temperature and Low Pressure Refrigeration EER

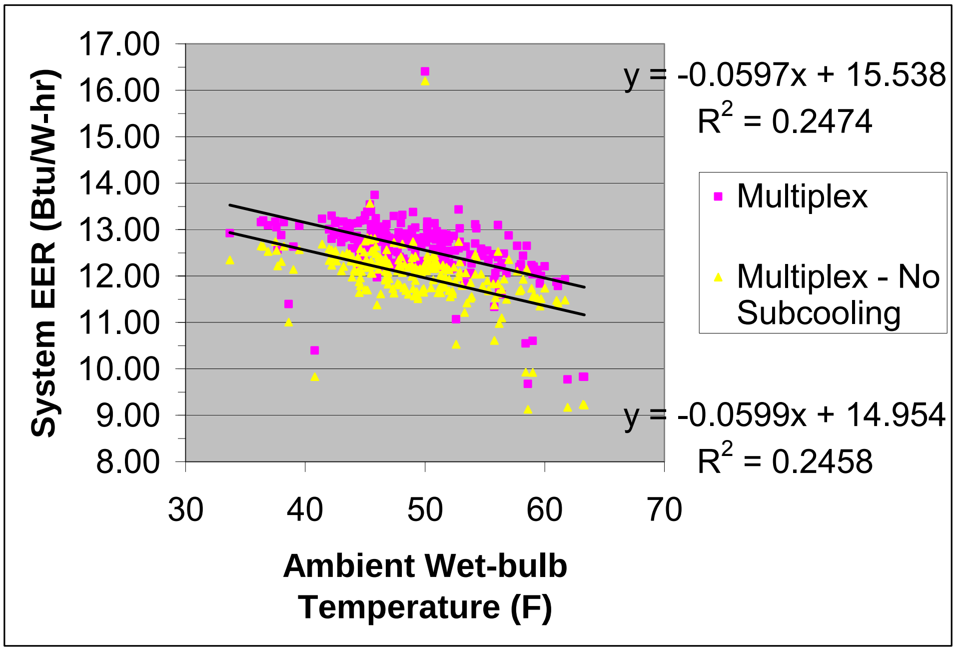

Figure 15. Relation between Ambient Wet-Bulb Temperature and Medium Temperature Refrigeration EER for the Multiplex System

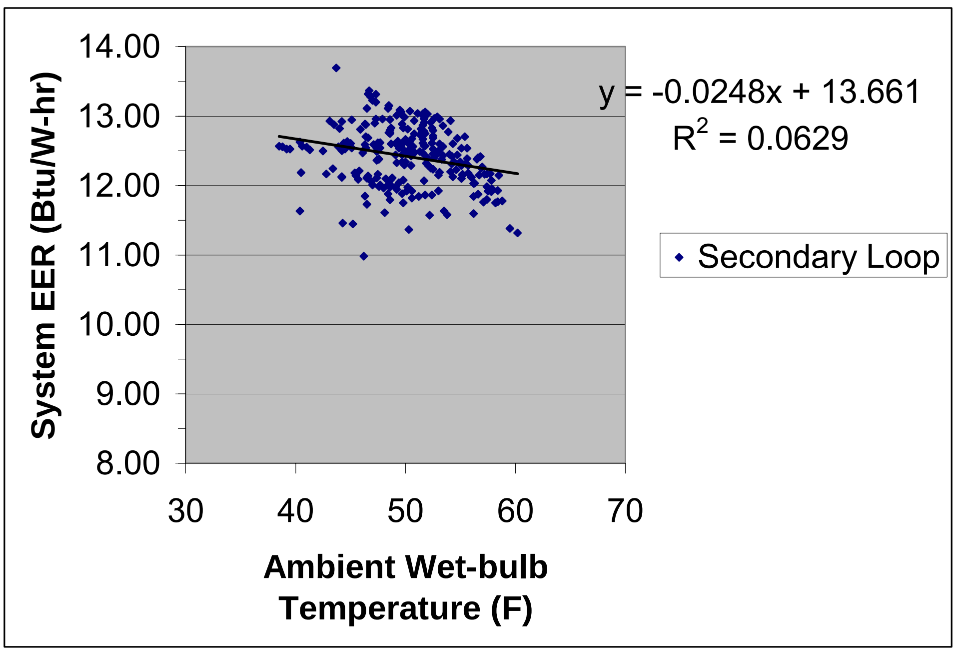

Figure 16. Relation between Ambient Wet-Bulb Temperature and Medium Temperature Refrigeration EER for the Secondary Loop System

3.2.4 Product Storage Temperatures

The ability of the two refrigeration systems to maintain product storage temperatures was evaluated. Two display case types were considered, which were:

- Single-deck meat
- Multi-deck produce

Cases of these types were available at the two test stores. These cases were equipped with product simulators that allowed the temperature of a thermal mass to be monitored.

Table 22 shows the average temperature measurements for the instrumented cases for a 24-hour period. The uniformity of the product temperature is characterized by the standard deviation (σ) of the product temperature. By definition, 95% of the product temperature values will fall within +/- 2σ. Temperature plots for the meat cases are shown in Figure 17 and Figure 18, for the multiplex and secondary loop stores, respectively. Similar plots for the produce cases are shown in Figure 19 and Figure 20.

Table 22. Comparison of Display Case Performance for the Multiplex and Secondary Loop Refrigeration Systems

	Compressor Rack SST	Discharge Air Temp	Product Temp	Standard Deviation in Product Temp
Single-deck Meat Display Case				
Multiplex	2.3	26.5	30.7	+/- 0.95
Secondary Loop	14.1	29.4	33.8	+/- 1.81
Multi-deck Produce Case				
Multiplex	21.0	36.0	37.7	+/- 1.35
Secondary Loop	20.9	37.4	37.1	+/- 0.88

For the single-deck meat cases, the product temperatures achieved by both systems are acceptable for storage. Both cases also operate at a similar temperature difference between the product and the discharge air – 4.2°F for multiplex and 4.3°F for the secondary loop. The multiplex system is operated at a significantly lower SST than the secondary loop system. It should also be noted from Figure 16 and Figure 17 the peak discharge air temperature of the secondary loop case is higher than the multiplex case. The maximum discharge air temperature for the secondary loop case was 56.4°F, while the maximum for the multiplex case was 42.4°F. The higher air temperature for the secondary loop case suggests that the defrost duration used was too long, causing the air and the product to warm up unnecessarily. With a proper defrost time setting, a more uniform product temperature can be expected.

The temperature results for the multi-deck produce cases were similar for both the multiplex and secondary loop systems. The SST of both racks was essentially the same. The temperature difference between the discharge air and the product was small for both systems. For the secondary loop case, the average discharge air temperature was slightly higher than the product temperature, which may be explained by an error in the temperature measurements and the air temperature includes values during defrost. The uniformity of the product temperature was better for the secondary loop produce case by a small amount.

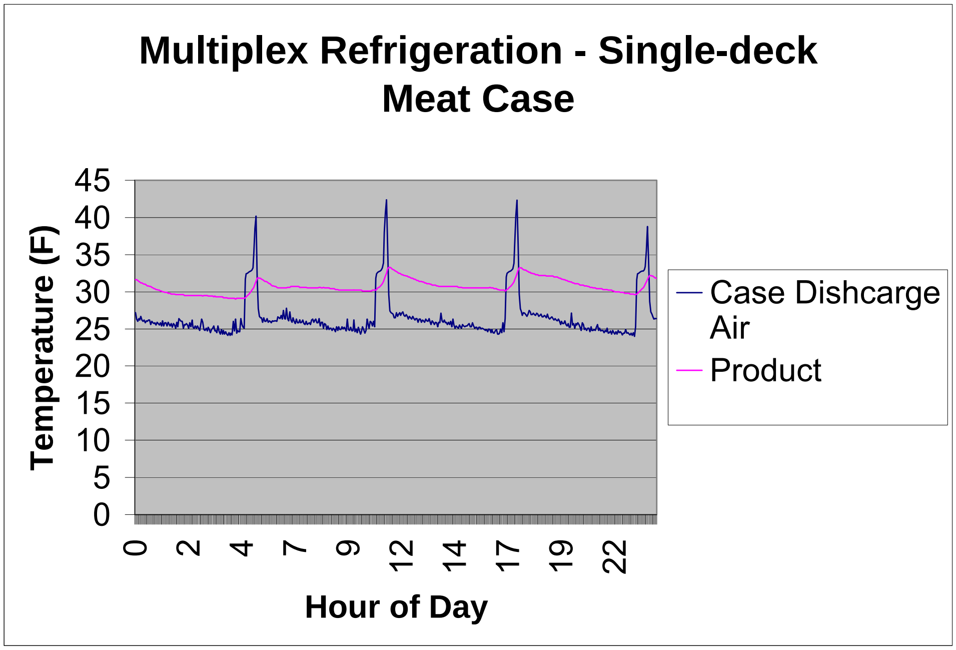

Figure 17. Discharge Air and Product Temperature Profiles for the Single-Deck Meat Case - Multiplex Refrigeration

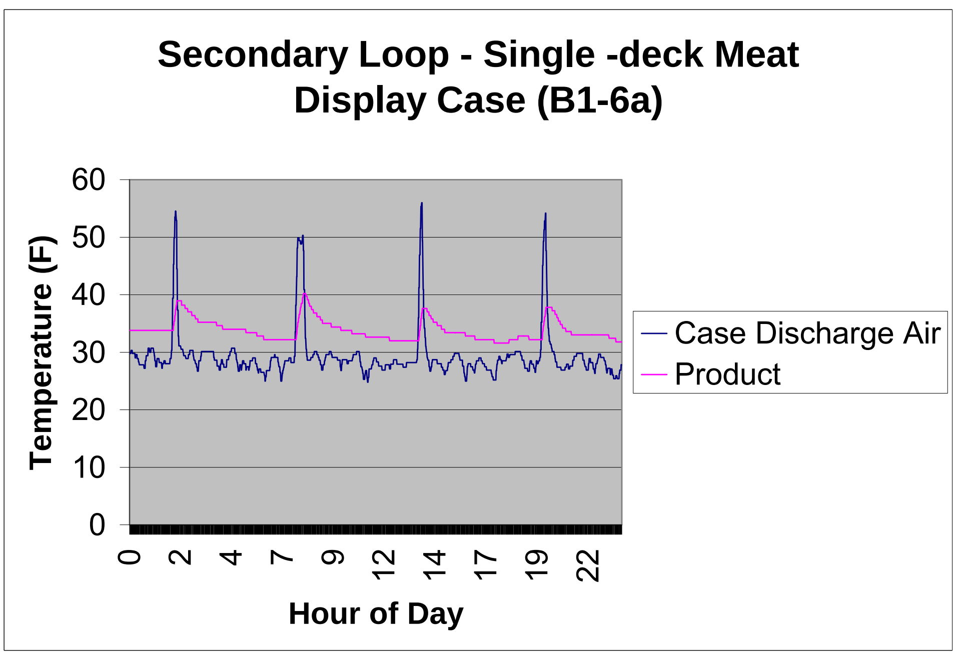

Figure 18. Discharge Air and Product Temperature Profiles for the Single-Deck Meat Case - Secondary Loop Refrigeration

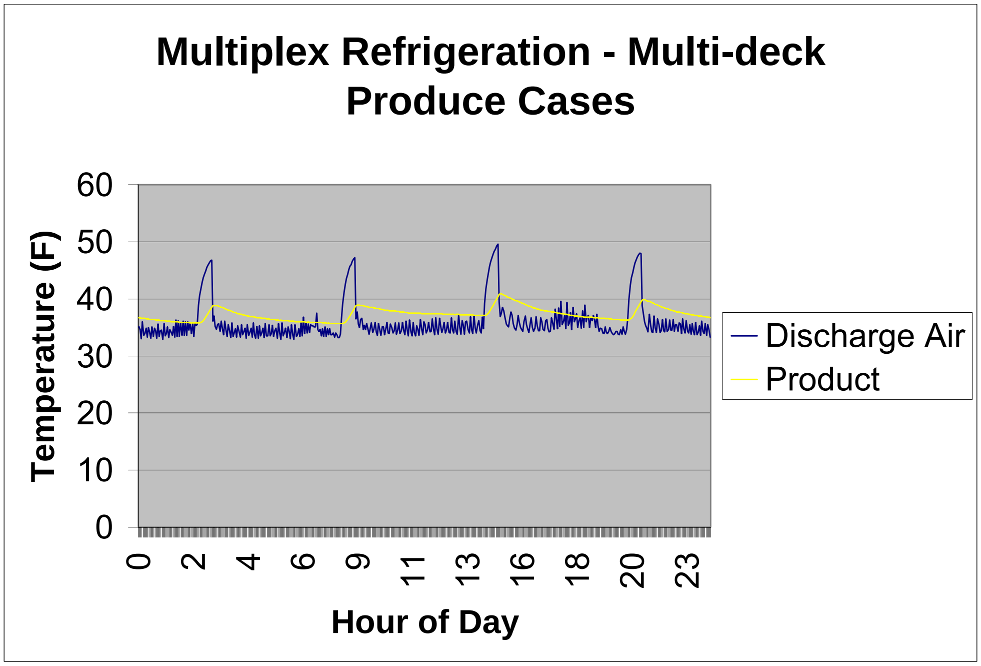

Figure 19. Discharge Air and Product Temperature Profiles for the Multi-Deck Product Case – Multiplex Refrigeration

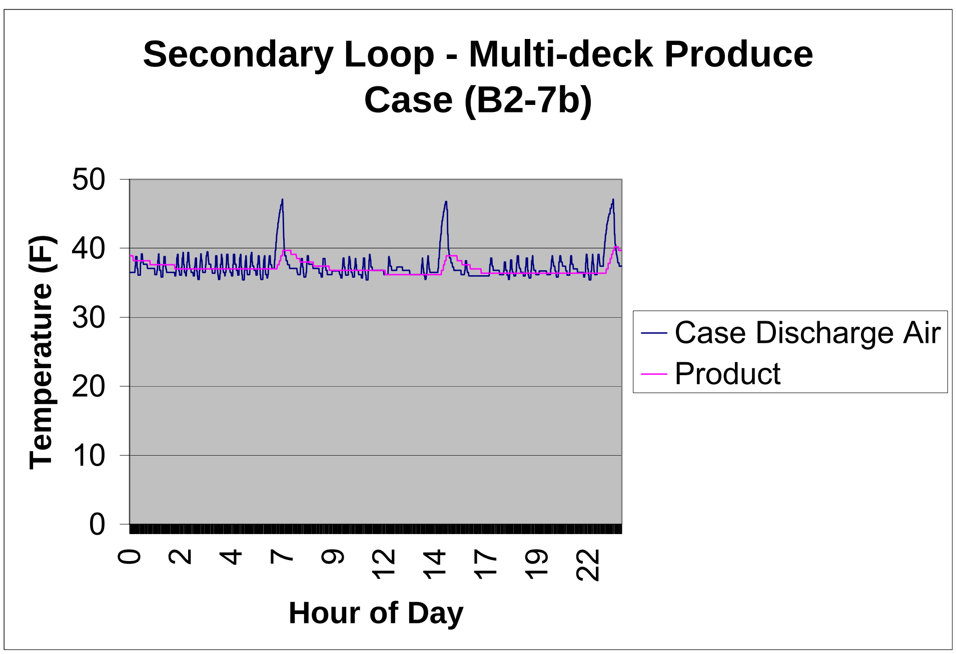

Figure 20. Discharge Air and Product Temperature Profiles for the Multi-Deck Product Case – Secondary Loop Refrigeration

3.2.5 Refrigerant Leakage Comparison

The refrigerant charge for the secondary loop refrigeration system was 1400 lb. This value is higher than the project goal of 500 lb or less. The added charge is primarily due to the of the additional refrigerant necessary to provide heat reclaim for both hot water and space heat. Heat reclaim is an effective use of the discharge heat generated by the refrigeration system which can replace a significant amount of energy in a supermarket.

The refrigerant charge for the multiplex system was not measured, but was estimated at 3,000 lb, or roughly 1,000 lb. per compressor rack.

Table 23 provides a record of the refrigerant additions at the two test stores. The information was obtained from Safeway, Inc for both stores. The refrigerant additions for the secondary loop store include 2, 100-lb additions that occurred in September 2002. These charges were required because of installation of flow meters in the refrigeration used for the field test work and should not be considered normal refrigerant loss by the system. The remaining 275 lb addition was due to a fitting break at the compressor rack. This one loss is believed to be the only recharge required by the secondary loop system since its start on December 18, 2001.

The records for the refrigerant charge addition for the multiplex store are very limited, because of a change in site maintenance contractor as of January 2003. Refrigerant records from the previous contractor were not available from Safeway, Inc. at the time of this report. The available information is very limited and may not be representative of the true refrigerant leakage of this multiplex refrigeration system. The total charge addition for the first four months of 2003 listed for the multiplex site is 72 lb.

Table 23. Refrigerant Additions at the Secondary Loop and Multiplex Stores

Vons 1610 -Secondary Loop Store			Vons 2030 - Multiplex Store		
Date	Refrigerant	Pounds	Date	Refrigerant	Pounds
9/1/2002	507	100	2/21/2003	404A	8
9/4/2002	507	100	3/14/2003	R22	60
4/18/2003	507	275	4/25/2003	R404A	4
Total		475			72

The annual refrigerant leak rate from each site can be determined using the methods provided by the USEPA (9). The percent of total refrigerant charge that is lost through leakage is found from:

$$\%\ \text{Annual Leak Rate} = \frac{\text{lb refrigerant added}}{\text{total refrigerant charge}} \times \frac{365\ \text{days}}{\text{days since last refrigerant additions}} \times 100$$

Table 24 shows the results for the two stores. With the exception of the addition of field test instrumentation, the only refrigerant charge addition credited to secondary loop store since the opening was a single recharge of 275 lb. The number of days between the store opening and the refrigerant addition was approximately 486 days. The limited records for the multiplex store suggest that the system was fully charged on February 21. Between that date and April 25, 64 lb of refrigerant was needed to maintain the system charge. The calculated annual % leak rates for the 2 stores were very close at 14.8 and 12.4 % for the secondary loop and multiplex stores, respectively, because of the large system charge associated with the multiplex refrigeration system. In terms of absolute refrigerant loss, the annual leak rate for the secondary loop store was less than that of the multiplex store by 44.3%.

Table 24. Estimated Annual Refrigerant Leak Rates for the Secondary Loop and Multiplex Test Stores

Store	Refrigerant Added (lb)	Days Since Last Addition	Total System Charge (lb)	% Annual Leak Rate	Annual Leak Rate (lb/yr)
Secondary Loop	275	486	1,400	14.8	206
Multiplex	64	63	3,000	12.4	370

4.0 Conclusion & Recommendations

The specific technical objectives of the project were to design and test a secondary loop refrigeration system, which:

- *Consumes approximately 14% less electricity than a state-of-the-art multiplex refrigeration system (baseline system) installed in a comparable store.*

The modeling results for the secondary loop refrigeration system configured for this project show annual energy savings of 145,894 kWh, or 14.9%, can be obtained when compared to the performance of a multiplex refrigeration system employing air-cooled condensing. This particular multiplex system configuration is the most common configuration installed in supermarkets and may be considered the baseline system. Modeling results also show that the savings were reduced to 6,130 kWh/yr when the secondary loop was compared to multiplex with evaporative condensing, suggesting that the largest portion of the savings can be credited to the use of evaporative condensing.

The field test results suggest the analytical results are in the correct direction. The savings achieved by the secondary loop refrigeration system versus the multiplex system with no medium temperature subcooling were 37,266 kWh/yr, or 4.9%.

The savings achieved by the secondary loop refrigeration system may be attributed to energy-saving features incorporated in its design. The annual savings achieved by each of these features compared to a more conventional secondary loop refrigeration system were:

- Multiple parallel brine pumps – Estimated annual savings versus single large pumps are 99,718 kWh.
- Subcooling from warm brine defrost – Estimated annual savings versus no subcooling are 49,570 kWh.

Additional savings were achieved by the use of a minimum delta T between the display case discharge air and refrigeration saturated suction temperatures, the use of a low-viscosity secondary fluid, and evaporative condensing. .

- *Has a refrigerant charge that is ten times less (less than 500 lbs.) than the baseline system.*

The advanced secondary loop refrigeration system as tested had an initial refrigerant charge of 1,400 lb., which is considerably larger than the stated goal. The reason for the larger refrigerant charge is the secondary loop system also had heat reclaim capabilities for both hot water and space heating. The added charge was needed for the additional piping and heat exchangers associated with heat reclaim. Heat reclaim is of great value to the operation of the supermarket since the heat reclaim can displace all energy use associated with hot water heating and space heating for the store.

- *Loses annually no more than 15% of its refrigerant charge due to leakage.*

Service records for the two stores indicated the refrigerant leak rate for the secondary loop store was on the order of 14.8%/yr, or 206 lb/yr. The multiplex system was estimated to have a leak rate of 12.4%/yr, or 370 lb/yr. The refrigerant data for the multiplex store were very limited because the maintenance contractor for the store had changed at the beginning of 2003 and a complete year of service records was not available. It is likely that a full year of data would show a significantly higher loss of refrigerant for the multiplex refrigeration system.

The ability of the two refrigeration systems to maintain product storage temperature was also assessed. The comparison for single-deck meat cases showed the multiplex and secondary loop systems maintained the product at acceptable temperature levels. The case associated with the multiplex system had a lower and more uniform product temperature than for the case operating in the secondary loop system. The multiplex display case had to operate a much lower rack SST in order to achieve this condition (2.3°F for multiplex vs. 14.1°F for secondary loop). For the multi-deck produce cases, the multiplex and secondary loop systems operated at similar rack SST values. The resulting average product temperature was approximately the same for both systems, but the product temperature of the multi-deck case in the secondary loop system was more uniform.

Benefits to California

This project contributed to the PIER program objective of reducing the environmental costs of California's electrical system, by developing an alternative refrigeration system which uses significantly less refrigerant than conventional systems. It also contributed to the PIER program objective of improving energy value of California's electricity by lowering electrical consumption of supermarket secondary loop refrigeration systems.

5.0 References

1. Sand, James R, Steven K. Fischer, Van D. Baxter*, Energy and Global Warming Impacts of HFC Refrigerants and Emerging Technologies,* Oak Ridge National Laboratory, sponsored by Alternative Fluorocarbons Environmental Acceptability Study (AFEAS), U.S. Department of Energy, 1997.

2. Terrell, Wilson, Y. Mao, and P. Hrnjak, *Tests of Supermarket Display Cases when Operating with Secondary Refrigerants*, International Conference on Ozone Protection Technologies, Baltimore, MD, November 12-13, 1997.

3. Cooper, P.J., *Experience with Secondary Loop Refrigeration Systems in European Supermarkets,* International Conference on Ozone Protection Technologies, Baltimore, MD, November 12-13, 1997.

4. Walker, D.H., Development and Demonstration of an Advanced Supermarket Refrigeration/HVAC System, Foster-Miller, Inc., Subcontract Number 62X-SX363C Oak Ridge National Laboratory, Oak Ridge, TN, September 2001.

5. Walker, D.H., and G.I. Deming, *Supermarket Refrigeration Modeling and Field Demonstration,* Foster-Miller, Inc., EPRI Report No. CU-6268, Electric Power Research Institute, Palo Alto, CA., March, 1989.

6. Kazachki, G., E. Bayoglu, and C. Gage, "Comparative Evaluation of Heat-Transfer Fluids for Secondary Loop Systems," presentation given at the International Conference on Ozone–Protection Technologies, Baltimore, MD, November 13, 1997.

7. Walker, D.H., Foster-Miller, Inc., *Field Testing of High-Efficiency Supermarket Refrigeration,* EPRI Report No. TR-100351, Electric Power Research Institute, Palo Alto, CA., December, 1992.

8. Walker, D.H., Foster-Miller, Inc., "Development of an Evaporative Condenser for Northern Climates," Report prepared for Niagara Blower Co., Buffalo, NY, April, 1997.

9. *Compliance Guidance For Industrial Process Refrigeration Leak Repair Regulations Under Section 608 of The Clean Air Act*, The Chemical Manufacturers Association and The Environmental Protection Agency, October 1995.